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01 January 2010 I Volume 112

CHASSIS and Model-based Algorithm Development

PISTON ROD VIBRATIONS in Damper Modules

DRIVER-FOCUSED Configuration of Vans

SIMULATION ENVIRONMENT for the Analysis of Different Hybrid Powertrain Configurations

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HYBRID STEERING for Reducing Fuel Consumption of Commercial Vehicles

IN SEARCH OF THE GOOD SOUND

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COVER STORY

IN SEARCH OF THE GOOD SOUND

4, 12 I Vehicle acoustics at all times have been a special discipline regarding physics, legislation and comfort. The necessity of constructing low-emission engines and the trend to more and more electrification assign engineers new tasks. Tenneco examined what influences the design of the exhaust manifold, the valve timing and the silencer have on the sound image. Rieter reports about concepts for engine compartment encapsulation.

THE FASCINATION OF TECHNOLOGY

Dear Reader,

Compared to a car, a magazine has a model life cycle of about six years. In view of the economic crisis, we briefly considered, at the beginning of 2009, whether to keep the old model for one more year, and then put all our effort into a relaunch.

Now, as the economy is starting to recover, we can present to you and our advertising customers a completely new ATZ. The new design reflects our basic conviction that nothing can replace a well-crafted trade journal. But, of course, it must also be in keeping with the reading habits of its time. To fulfil these requirements, we have made a number of changes, which I'm sure you will notice.

: The overall appearance has become much "cleaner", with more white space and attractive graphics that have a new, contemporary design.
: Our cover story topic has now been given more space to develop – making use of the benefits of a printed magazine, which is able to examine an issue in depth. After all, there is enough sound-bite journalism to be found elsewhere, especially on the internet.
: Our interviews now have an extra page to allow our interview partners to present their ideas in more detail.
: In the Industry section, the fascination of technology that motivates all of us is now presented even more effectively in larger, more powerful images.
: The Science section is now more clearly set apart from the remaining contents. It is characterised by formal precision and a clear design.

At a stroke, ATZ has become more modern. Throughout the 112-year history of this famous magazine, we have always succeeded in adapting to changing times while at the same time remaining true to ourselves, and particularly to the technical-scientific orientation for which this magazine is renowned. Visual appearance is not everything. We need to keep our finger on the pulse of technology and to ask the best authors to address the most important challenges in automotive engineering.

I look forward to your feedback, and especially your response to our new model generation.

JOHANNES WINTERHAGEN, Editor-in-Chief
Wiesbaden, 30 November 2009

TAILPIPE SOUND QUALITY IN A THREE-CYLINDER ENGINE

Three-cylinder engines are an acoustic challenge, but if the exhaust system is well designed, they can convey a sound image appropriate to a luxury car: generally quiet and without dominating orders. Using simulation processes and test stand testing, Tenneco examined what influences the design of the exhaust manifold, the valve timing and the silencer have on the sound image.

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PERCEPTION OF THE SOUND IMAGE

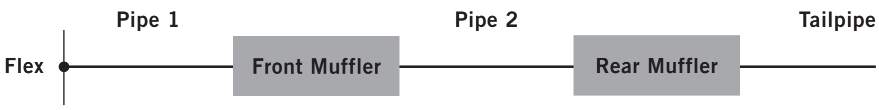
Car makers now require a particular sound pattern for a car or a vehicle platform. In this case, the tailpipe noise of the exhaust system plays an important role in the subjective impression of the car [1]. Other design challenges include reductions in fuel consumption and emissions. Increasing numbers of midsized cars are therefore now using turbocharged three-cylinder engines [3; 4], although these do not offer the same sound quality as four- or six-cylinder engines. Most people perceive a three-cylinder engine as having a poor sound quality with a rough character and lacking in performance. Exhaust systems suppliers are therefore faced with the task of developing technical solutions that provide the three-cylinder engine with a positive sound image.

First of all, a benchmark test was conducted. The measurements showed that three-cylinder engines have many different sound colorations, from the very sporty to quiet and luxury saloon-like. As simple psychoacoustic analyses are not sufficient to rate the sound coloration, in the following we describe a listening model that is able to provide the tonal component “emergent sound pressure levels” (SPL). As objective and subjective data cannot provide a real listening experience of the sound, we also present an original sound simulation tool. These tools are used to evaluate the influence of exhaust design modifications on sound coloration.

The first step is to analyse the exhaust system hot-end design [5]. The hot end describes the exhaust parts that are located close to the engine, and generally includes the manifolds, downpipes and catalytic converters. It has already been demonstrated that exhaust hot-end modifications are a good way of reaching this goal without drastically affecting the engine performance [6]. However, cold-end modifications – the cold end includes the exhaust parts from the flex element to the tailpipe outlet – do not have dramatic effects on the low engine orders levels but they do enable fine tuning of the tailpipe timbre and, above all, influence the flow noise character [7].

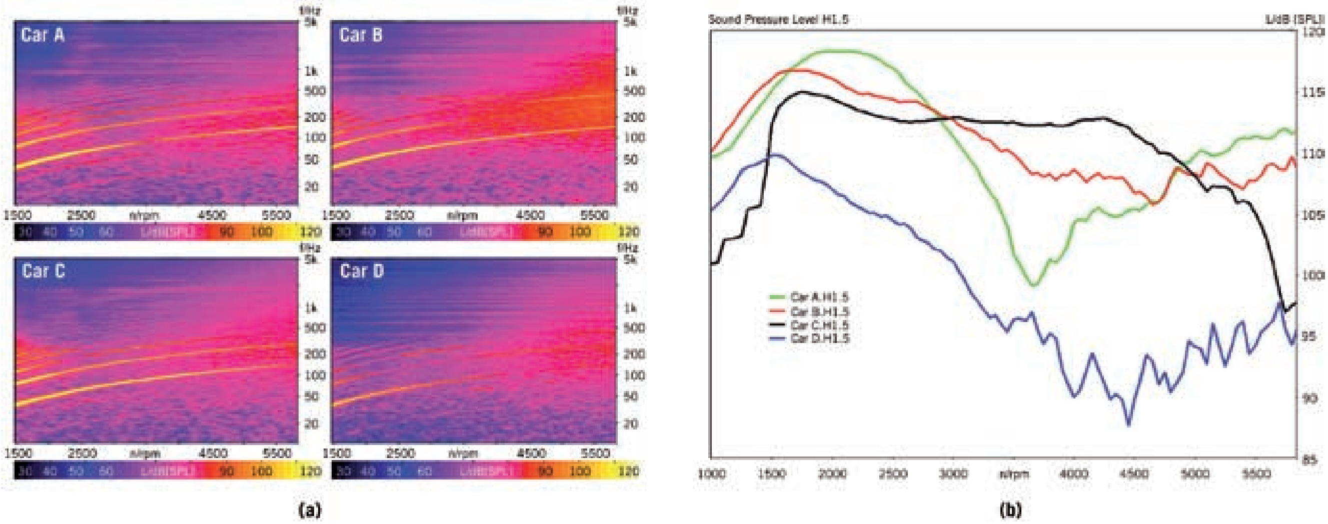
A benchmark test was first conducted with four cars, ❶. Cars A and C had the same engine, a 1.0 l with a rated output of 50 kW. Car B had the same displacement but an output of 44 kW, while car D had a 1.2 l engine, also with 44 kW.

❷ shows the Campbell diagrams (auto-power spectrum as a function of engine speed per minute-rpm range) of the tailpipe noise during run-ups. The sound coloration differs strongly between car D and the three others. The sound of Car D exhibits lower levels of engine order noise. Its sound coloration has a luxury saloon-like character with some pleasant flow noise in the medium rpm range. Car A exhibits a loss of sound at around 3500 rpm, with the result that the timbre variation is too great to be favourably rated: it does not have a luxury saloon-like character. Car B exhibits a badly rated high flow

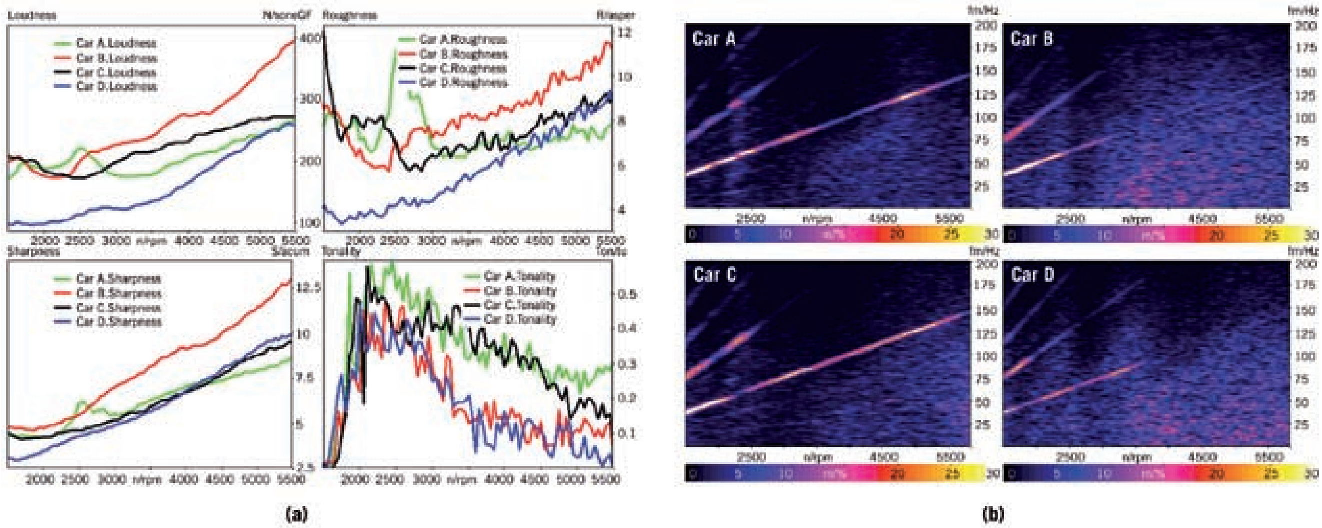


	CAR A	CAR B	CAR C	CAR D
PIPE 1, LENGTH (MM)	1200	350	3000	920
PIPE 1, DIAMETER (MM)	35	45	35	40
FRONT SILENCER VOLUME (L)	2.5	3.8	not available	3.5
PIPE 2, LENGTH (MM)	1700	2500	not available	850
PIPE 2, DIAMETER (MM)	35	38	not available	40
REAR SILENCER VOLUME (L)	6.6	7.5	5.2	10.8
TAILPIPE LENGTH (MM)	160	300	350	950
TAILPIPE DIAMETER (MM)	39	38	35	40

❶ Main geometrical data of the exhaust systems examined



❷ Measured tailpipe noises: APS versus rpm range (a), SPL of the H1.5 Engine order (b)



❸ Tailpipe noise: psychoacoustic parameters (a), modulation frequencies 180 Hz octave band (b)

noise level at the end, but its engine order sound is loud enough to be considered as a small sporty car. Car C has a sporty sound character at the beginning due to the presence of a large number of engine orders. The flow noise level is also at a minimum level. For all cars, one can also observe that low engines orders such as H0.5 and/or H1 may be present. ❸ shows the psychoacoustic character of each tailpipe sound. The loudness measurement confirms that car D is the quietest car. The other three cars are about the same, except for at the end of the run-up process, when flow noise levels in car B drive the loudness higher. With regard to sharpness and tonality, all cars can be considered as almost identical. The biggest difference is found when roughness is measured. Car A exhibits high roughness at around 2750 rpm, exactly when the level of engine orders fall. Indeed, careful listening can detect a rasping noise at this rpm value.

As reported in [8], a flutter noise pattern appears at the beginning of WOT run-ups. The flutter noise can be verbally described as a typical “horse breath” pattern. It relates to a narrow-band high-level noise with a periodically modulated amplitude, ③, right. This phenomenon is due to a high-pressure flow noise modulated by the engine orders. The ear detects only the modulation effect obtained by the lowest engine orders. In the H1.5 SPL, high levels were observed between 1500 rpm and 2500 rpm, thus increasing this flutter noise sensation. Consequently, a high loudness was perceived; this sound had no particular tonal or periodical property.

As the engine power of a three-cylinder naturally aspirated engine is too low for the use of this kind of engine in the midsize segment, turbocharged engines will be used more frequently in the future. ❹ displays some psychoacoustic parameters obtained for tailpipe sound measurements. The vehicle with a turbocharger has a better sound quality than the naturally aspirated one if the specification requires a luxury saloon-like character. The sound coloration varied slightly throughout the entire run-up. Moreover, the turbocharger seemed to attenuate the bad character of the sound without increasing the loudness sensation. From the tailpipe noise point of view [9], the turbocharger is very useful in reducing annoying high-frequency noise occurrences [10].

THE LISTENING MODEL AND ITS APPLICATION

In the literature, the term psychoacoustic model refers to two ways of describing the listening sensation mathematically. The first category of psychoacoustic models aims at reproducing the way in which the ear analyses incoming acoustic signals. They can be considered as human signal processing tools. The second category refers to the establishment of sound quality models that are mainly based on statistical results provided by listening clinics [11]. A recent article [12] proposes an interesting equation to compute the "emergent SPL" of aurally relevant tonal components. In this case, the tonal components are the engine orders. The "emergent SPL" is given by:

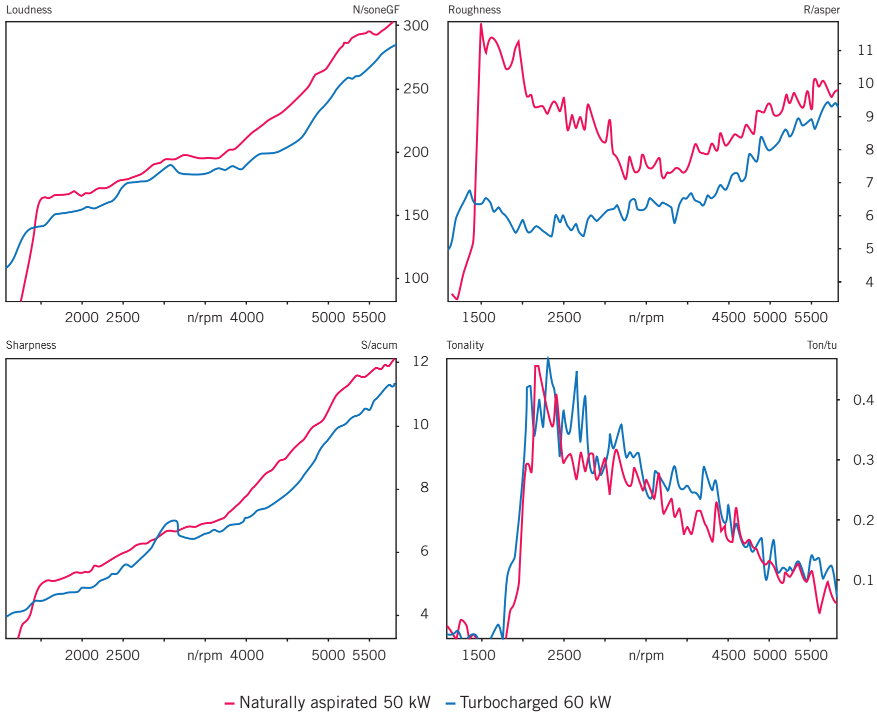
EQ.

$$xL_i = L_i - 10\log\left[\left(\sum_{j=1\neq i}^{N} 10^{L_{Ej}/20}\right)^2 + I_i + 10^{L_{THi}/10}\right]$$

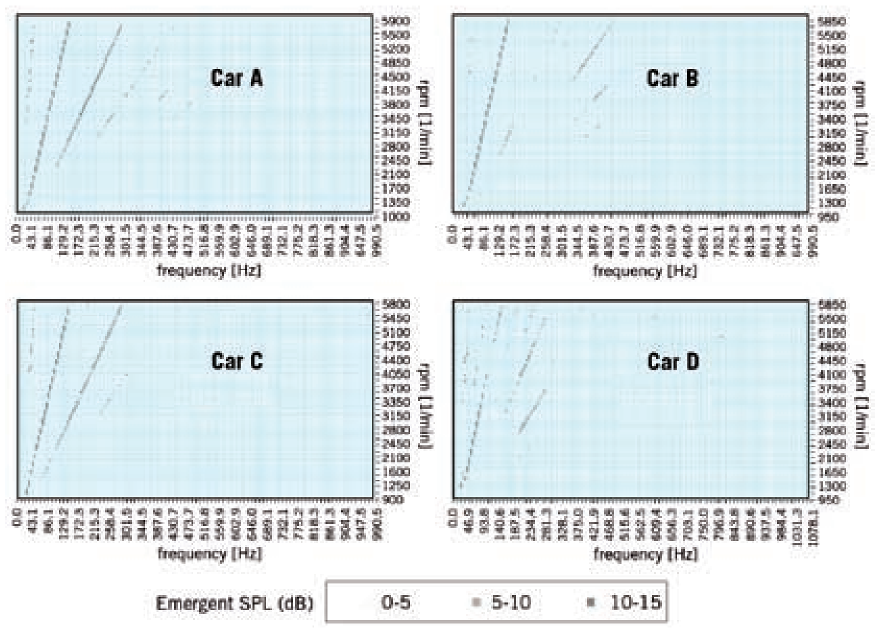
where L_i is the ith SPL of the tonal component and xL_i the "emergent SPL", L_{Ej} is the excitation level of the jth masker tonal component, I_i is the background noise intensity in the critical band, whose centre frequency is equal to the ith tonal component, and L_{THi} is the hearing threshold SPL at the ith location. The masking threshold curves are the most difficult data to compute as, except in the case of the MPEG Layer 3 audio encoder, there are no standardized formulas.

The results are displayed in ❺. Cars A and C have a high tonal character with an acoustic energy that is well located and balanced between H1.5 and H3. The ear also perceives some energy due to the H0.5 order. As the "emergent SPL" is around 15 dB, the tonality property of these cars is quite high, which explains some of the sporty coloration. As the two tones have a high "emergent SPL", the sound was not judged as boring. However, the cars were not rated as highly sporty because the sound was not rough enough.

The comparison between results from car B and car D is very interesting. Car B exhibits a one-tone sound coloration; the H1.5 engine order is dominant. Sometimes, acoustic energy also appears due to the H4.5 engine order. This variation is not really perceived. This kind of sound is often judged as boring and is sometimes poorly rated. Car D was described as a good luxury saloon-like example of a 3-cylinder application. There is no really dominant engine order and even when some variations were detected, they were not perceived by an



❹ Tailpipe noise: comparison between a naturally aspirated and a turbocharged engine

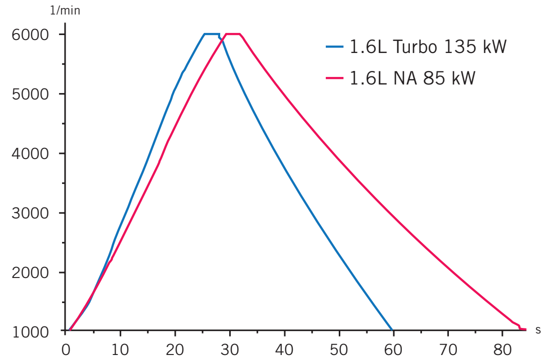


❺ Emergent SPL of the engine orders provided by the listening model expert team. As the tonality index given in ③, left, is the lowest, one can conclude that this sound has less tonal character than the others. These two properties define the luxury saloon-like character well.

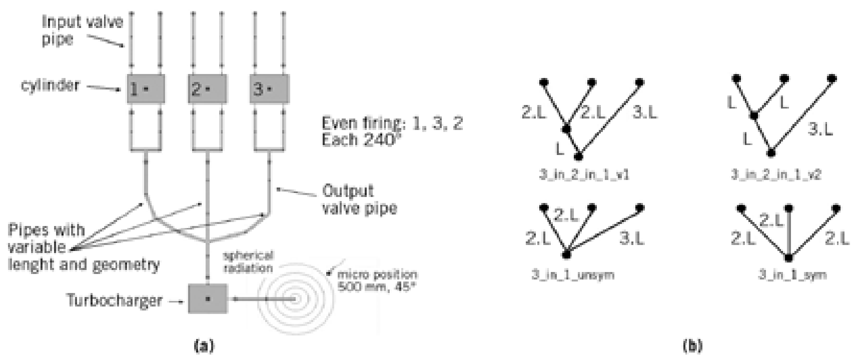
SYNTHETIC TAILPIPE SOUND SIMULATION

The sound simulation tool presented in the following is based on the computational results obtained from acoustic and fluid mechanics software. The main focus is on the tailpipe sound simulation of a WOT run-up. Run-up tailpipe sounds are composed of three components: the engine order sound resulting from the combustion process, the flow noise resulting from aerodynamics sources due to the exhaust gas flow disturbed by solid discontinuities, and flutter noise, an interacting component between engine orders and flow noise that appears at low engine speeds during WOT run-ups. This latter is fundamental in obtaining a realistic synthetic noise and demonstrates that the tailpipe sound is not just an addition of both of the first two components. Engine orders are tonal components. It seemed natural [14, 15] to model each engine order by a swept sine function in which the swept function is defined by the engine rpm time history during WOT run-up. ❻ displays the rpm time history measured on a chassis dyno for a 1.6 l petrol engine with and without a turbocharger. Each engine order SPL was adjusted by using rpm-variable order level filtering.

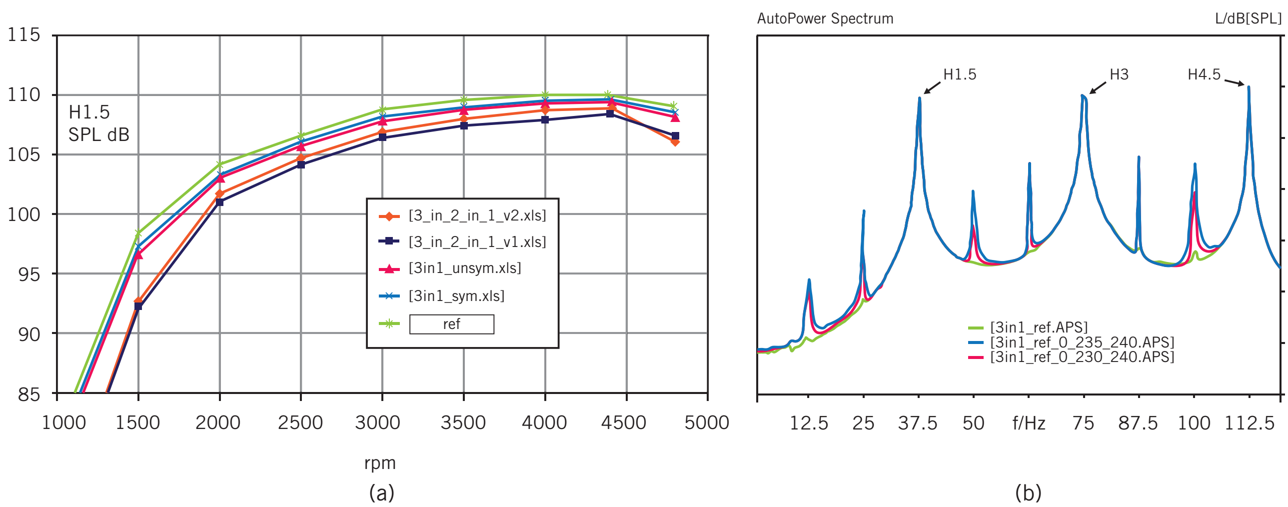
The flow noise component is certainly the most difficult one to predict as it first requires the computation of the unsteady exhaust gas flow. Its frequency content is strongly coloured by the pipe resonances. The easiest way to create a semi-realistic flow noise is to use a white noise filtered by a FIR filter whose transfer function varies with time (or rpm). Pipe resonances were used to build the FIR filter transfer curve. This sound pattern is the result of an interaction between flow noise and engine orders sound. At Tenneco, this flutter noise was modelled as a coloured low-frequency noise whose



❻ Rpm time history measured on a chassis dyno bench during WOT run-up



❼ Overview of the GT Power Model (a) with the four manifold geometries investigated (b)



❽ SPL of the H1.5 engine order for the manifolds (a) and APS at 1500 rpm when valve timing modifications are implemented (b) amplitude is modulated by the lowest engine order.

MODIFICATIONS TO THE EXHAUST SYSTEM

In the following, the modifications to a three-cylinder turbocharged diesel engine are described, ❼. Only the engine, the manifold and the turbocharger have been taken into account. Both valve timing modifications and manifold geometrical changes were performed. Part b of ⑦ shows four of the five different manifold geometries investigated. The fifth case was the original one, which was considered as the reference system.

Valve timing manipulation is a good way of modifying the sound coloration of the tailpipe sound by amplifying or reducing the SPL of the engine orders [12]. The engine performance is also changed, but not by more than 2.5 %. The valve timing modification was implemented on only one cylinder and the manifold geometry was that of the reference case. Three configurations were investigated: no modification, 0_235_240 (a delay of 5°) and 0_230_240 (a delay of 10°). ❽ displays the Auto Power Spectrum (APS) computed at the tailpipe. The main engine order levels (H1.5, H3, H4.5) were not affected by the valve timing modification, whereas the SPL of the neighbouring engine orders strongly increased, resulting in a change in the sound timbre, ❾.

	H1.5	H3	H4.5	OTHER
BASE VALVE TIMING	10	18	7	not audible
0_235_240 CONFIGURATION	7.5	15	5	not audible
0_230_240 CONFIGURATION	5.5	12.5	3	not audible

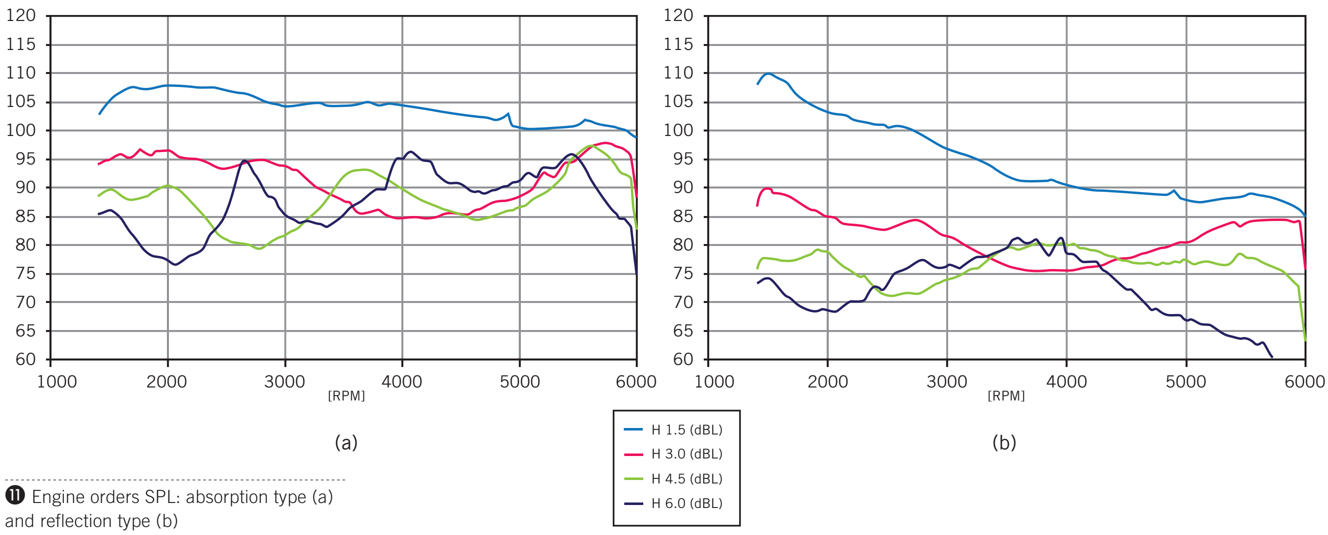
❾ Emergent SPL (dB) with valve timing modification but without manifold geometry changes

	H1,5	H3	H4,5	OTHER
3INTO2INTO1_V1	2.6	16	5.2	not audible
3INTO2INTO1_V2	3.2	16.,3	6	not audible
REFERENCE	7.8	19	6.7	not audible
3INTO1_SYM	8	19.3	7	not audible
3INTO1_ASSYM	7.5	19.2	6.7	not audible

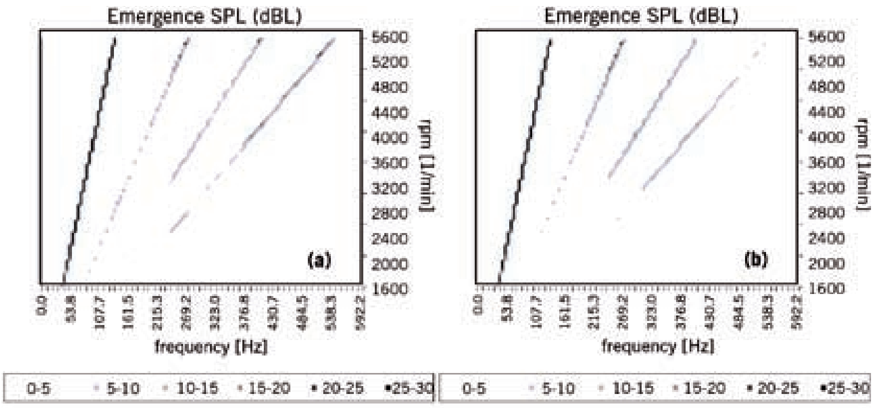
❿ Emergent SPL (dB) with manifold geometry changes but without valve timing modification

The studies carried out at Tenneco now focused on a change to the manifold geometry with unchanged valve timing. ⑧ shows the H1.5 engine order level for a WOT run-up obtained for each of the manifold geometries. Two classes of manifold geometries were distinguished: 3_into_2_into_1 and 3_into_1. The first class provided a high reduction in the order level by up to -6 dB. The other class had almost an equivalent order level within a 2 dB range. If one accepts a minor decrease in engine performance, the sound coloration is strongly improved, ❿. Only a manifold modification had a noticeable influence on the main engine orders levels. The valve timing modification was merely used for fine tuning the sound.

Two classes of exhaust cold-end designs can be distinguished depending on the acoustic damping mechanism of the silencer: absorption-type silencers and reflection type silencers. Except for the cold-end design, everything was kept identical: the engine, intake system and hot-end exhaust system. The cold-end absorption type was composed of two absorption silencer boxes with volumes of 6.1 and 6.8 l. The cold-end reflection-type silencer was composed of two silencer boxes: the closest to the engine was a purely absorption type (4 l) and the rear silencer was a purely reflection type (12 l). In both cases, the silencers were located at the same position. ⓫ shows the main engine order levels obtained in each



⓫ Engine orders SPL: absorption type (a) and reflection type (b)



⓬ Listening model results: absorption type (a) and reflection type (b)

configuration. If one examines the curves, one can expect a very different sound coloration. ⓬ displays the listening model results. First of all, the H1.5 was obviously the dominant order in both cases, but the sound was never perceived as a monotonous sound. Timbre variations were heard but the variations were felt differently depending on the exhaust system cold-end design. In the absorption case, strong timbre variations were perceived throughout the run-up. In the reflection case, the sound variations were smoother: the H1.5 order first dominates and then the H3 was perceived with a slowly increasing emergent level. Then, above 3200 rpm, the H4.5 and H6 were also perceived. This sound corresponds to the desired luxury saloon-like character. Therefore, one can conclude that the listening model is an efficient tool for balancing tonal loudness.

CONCLUSION

First of all, the benchmark provided very useful new information. It is interesting to notice that a luxury saloon-like character can be achieved without too much difficulty merely by using two different silencers. This sound character is the one most frequently used for midsize cars. A sporty coloration can be easily achieved by increasing the roughness thanks to the presence of H0.5 amplitude modulation, especially in the mid-rpm range. The use of absorption-type silencers is absolutely essential in this case. Finally, the listening model provides interesting information that allows a better understanding of the influence of the loudness balance between the firing engine orders on the engine sound.

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INNOVATIVE CONCEPTS FOR THERMO-ACOUSTIC ENGINE COMPARTMENT ENCAPSULATION

There is a clear trend towards retaining heat in the engine compartment for as long as possible in order to benefit fuel consumption. At the same, it is becoming increasingly difficult to achieve a quiet and pleasant exterior noise, because more efficient combustion is usually noisier combustion. According to Rieter and Nissan, both problems can be solved with clever encapsulation that, at an acceptable cost, cuts CO_2 emissions by 2.5 g/km and reduces exterior noise by up to 5 dB(A).

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CONTRASTING REQUIREMENTS

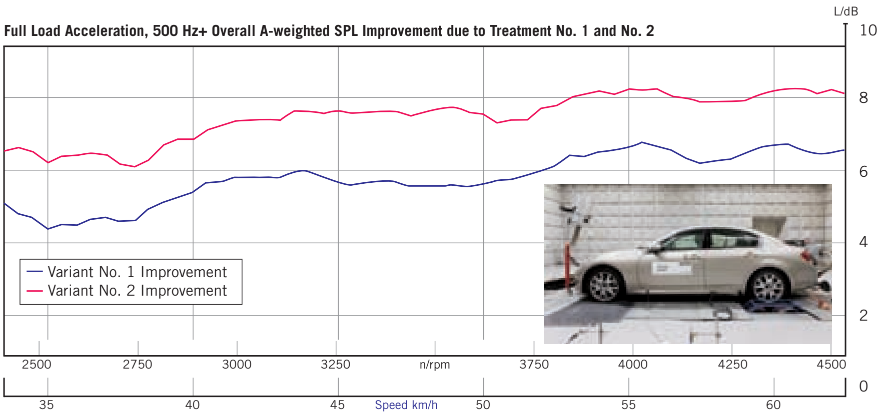
The need to further reduce the noise of the engine has always been in conflict with the requirements of thermal safety and the necessity to reduce weight and cost. This conflict could become even greater in future, since the engine compartment of future conventional and hybridised vehicles will need to be more enclosed than today. The main reasons for this are the tighter regulations envisaged for exterior noise, the ever-growing requirements for exterior and interior noise quality and new European CO_2 legislation, all of which result in higher mean effective pressures and therefore tend to make engines noisier. In addition, the aim is to store heat in the engine compartment for as long as possible in order to reduce fuel consumption after a cold start. A further objective is to optimise the aerodynamic flow in the engine compartment.

The encapsulation of the engine compartment arises from the need to reduce exterior noise, which is regulated by law. Typically, it consists of components fixed to the body. One exception is the engine top cover, which is attached directly to the engine. Acoustic absorbers are applied to the hood, the bulkhead, the sides of the front beams and an undershield. Today, undershields are partially covered by an acoustic material or made of an intrinsically absorbing and at the same time structural fibre-based material. Another trend is engine-mounted acoustic insulation components, most typically on the oil pan and in the area around the fuel injectors. The advantages of such components are that they treat noise directly at the source and have a reduced size and therefore weight. However, a challenge for materials in contact with the engine is that they are exposed to strong vibrations, high temperatures and aggressive chemicals. Furthermore, there is the difficulty linked with interfacing the shields with a very complex geometrical environment.

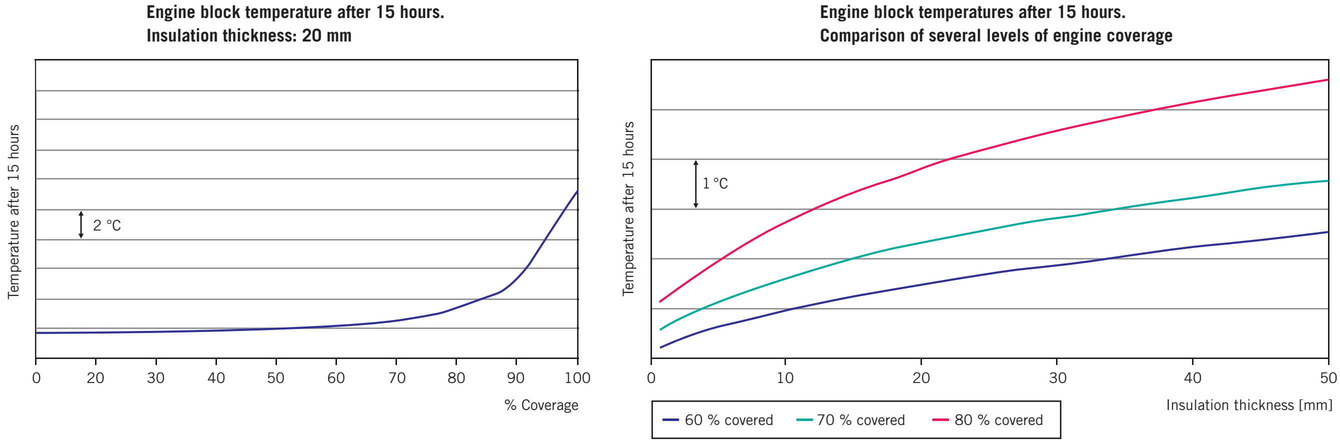
REDUCING BOTH FUEL CONSUMPTION AND EXTERIOR NOISE

Several studies [1] conducted by Rieter have shown that efficient encapsulation can provide a reduction in CO_2 emissions by 2.5 g/km in the New European Driving Cycle (NEDC). At the same time, it was demonstrated that underbody panelling can make a significant contribution towards reducing drag [2].

Both measures will be required in the future, as not only CO_2 but also noise emissions will be more strictly limited by EU legislation. Transport noise in the EU is to be reduced by up to 10 dB(A) by 2020 [3] through improvements to vehicles and infrastructure. The ongoing introduction of the new exterior noise test method might account for about a 2 dB(A) reduction in the reported noise values. However, there is still a lot to be done. Some OEMs expect that a further 3 dB(A) reduction in vehicle noise emissions will be needed.



❶ Average noise reduction for two different encapsulation approaches and exterior noise measurement of the powertrain (exhaust, tyre, intake masked) of the Nissan Infiniti in the semi-anechoic room



❷ Temperature of the engine block after 15 hours of cooling, as a function of the thickness of the encapsulation (left) and of the area coverage (right)

Several of the engine-based measures needed to improve fuel efficiency (such as higher injection pressure, direct petrol injection, downscaling to three or even two cylinders, turbocharging) also have a negative effect on the radiated noise. Tests on a vehicle with a 2 l petrol engine [4] show a potential of a maximum 1.5 dB reduction in exterior noise by optimising the existing body-mounted parts. In 2007, a system of engine-mounted parts weighing less than 2.5 kg was presented [5]. The system provided reductions in radiated noise in the engine test bench of 3 to 4 dB. Equipping parts of the engine air-intake system with porous acoustic materials integrated into a double-shell panel provides significant improvements in the exterior noise quality of a diesel vehicle [6]. The most recent study conducted on a Nissan Infiniti with a 3.7 l V6 petrol engine shows that body-mounted engine encapsulation can reduce powertrain noise by 5 dB, ❶. Overall, the studies conclude that a combination of material optimisation and body-mounted and engine-mounted elements will be needed to achieve the legal requirements for significant noise reduction.

CONCEPTS FOR THERMO-ACOUSTIC ENGINE ENCAPSULATION

Body-mounted encapsulation is made of components originating from acoustic treatment: the bonnet absorber, outer bulkhead and under-engine shield. New elements are the vertical elements along the front beams and connected to the above-mentioned elements in order to form a tightly enclosed engine compartment. It is also necessary to use electrically actuated shutters or other appropriate front closing elements to limit heat and noise leakage through the radiator. The system must ensure an efficient flow of air through the radiator for optimum engine cooling during driving. It should also block the dissipation of warm air when the vehicle is parked. Since vibrations and operating temperatures are lower for body-mounted components, lightweight materials such as foams and felts can be used. On the other hand, the surface to be treated with thermo-acoustic materials is greater in comparison to the engine-mounted system and cost-effective development of the sealing system against the bonnet can be challenging. Several pass-throughs for the steering column, drive shafts, cables and other interfaces need to be integrated, allowing easy assembly.

Engine-mounted encapsulation is smaller but is fixed to surfaces that are usually hot and exposed to more vibration than the body. This requires more stable materials with higher temperature resistance. The subdivision of the encapsulation into its elements must allow access to the powertrain for maintenance. In practice, it is not possible to cover 100 % of the powertrain surface. Some ancillaries such as the alternator or the fuel injectors need to be cooled by air. Coverage of 85 % is, however, the minimum needed to achieve effective thermal encapsulation.

7 mm thick encapsulation covering 80 % of the area is equivalent to 50 mm thick encapsulation covering only 60 %, ❷. Therefore, the first priority in the development of an encapsulation is the highest possible degree of area coverage rather than the choice of highly insulating materials.

The potential of both approaches was evaluated in several vehicle tests, ❸. It was shown that engine-mounted encapsulation has a slightly lower heat storage capacity, although one needs to consider that this design was more difficult to realise on the existing engine due to lack of space. In new engine developments, the possibility of increasing the area coverage and the average thickness of the insulating materials improves significantly.

In the case of a body-mounted concept, it is usually easier to obtain sufficient area coverage, but the sealing of the system is more difficult. The weight of currently used systems for the thermal encapsulation of a four-cylinder engine is estimated at a maximum of 8 kg. For a B-segment car with a diesel engine, 6 °C higher powertrain temperatures in the NEDC translate into a CO_2 reduction of about 3 g/km, ❹. Considering the additional weight of the encapsulation, the actual CO_2 reduction can be calculated as 2.5 g/km. To achieve a comparable reduction by means of vehicle weight reduction, the weight would have to be reduced by 40 kg. The financial assessment of this advantage should also consider the new CO_2 limit values, which foresee monetary fines for fleets that exceed the limits.

One of the challenges in the implementation of thermo-acoustic encapsulation is the fulfilment of the thermal safety requirements under high load. This is in conflict with the wish to retain as much heat as possible when the vehicle is not in use. In body-mounted encapsulation, either the air coming from the radiator or the airflow from a special duct can be used to flush hot spots. If the encapsulation is also meant to provide improvements in aerodynamics, the airflow through the engine compartment should be minimised. A flow analysis of the engine compartment via CFD (computational fluid dynamics) and heat radiation computations show appropriate solutions in the concept phase.

COMPONENT ARCHITECTURE

In order to maximise the benefit of the overall thermo-acoustic system, Rieter has focused in its product development on minimising weight and the interactions among components in the system. Packaging space is limited and assembly and service requirements further increase the complexity of the whole system.

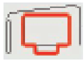
The Rie-BAY panel developed by Rieter, ❺, replaces several covers and the bonnet absorber by a single element, thus allowing a significant weight saving. Despite the full area coverage for heat storage and noise insulation, thanks to a new material based on temperature-resistant fibre-based materials, the solution leads to a weight advantage of about 50 % compared to injection-moulded parts, ❻. The construction comprises a lower and an upper shell made of function-specific, structurally and acoustically optimised composite materials. The double-shell design efficiently combines the following:

- high noise absorption
- high part stiffness
- heat insulation
- integration of cooling air ducts or engine air filtration and inlet
- integration of pedestrian protection absorbers, electric cables and devices.

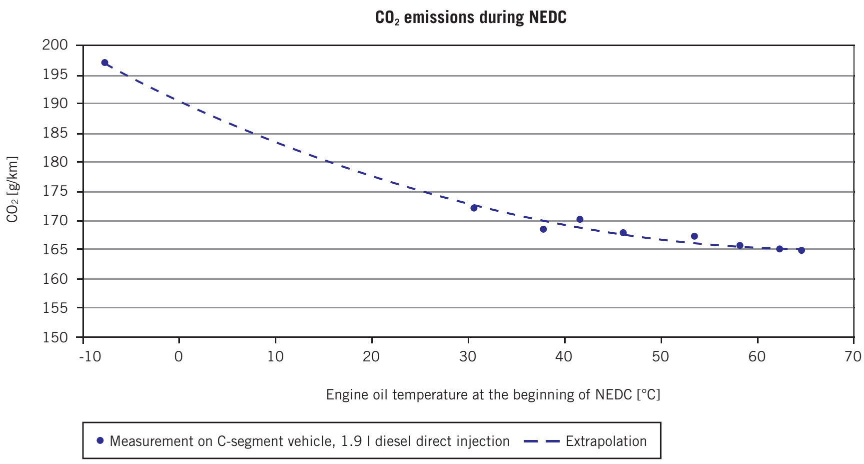
The requirements of the OEM with regard to haptics and optical appearance are fulfilled by the use of various foils or textile surfaces. Logos and colour surfaces can also be integrated into the upper shell. At the engine side, textile materials are used to obtain the required acoustic parameters. The use of heat-reflective foils outside or inside the shells provides an additional 1 °C of residual heat in the cool-down process. The component edges are designed in such a way that the sealing function towards the bonnet can be integrated in order to replace any heavy rubber seals. This results in another significant reduction in weight and assembly time. Design and quality control benefit from a leaner tolerance management with a single part when compared to the parts set used today. The patented double-shell principle also allows the compact integration of air ducts directly into the components. Further possible integrations could include electronics, electric cables and air filters. The Rie-BAY panel offers an almost package space neutral enhancement for heat and noise encapsulation, which can be used relatively independently of the engine variant installed in the car model. An additional engine top cover is eliminated by the proposed solution. Finally, the material concentration into one single module also improves end-of-life recycling.

MATERIALS FOR THERMAL AND ACOUSTIC INSULATION IN THE ENGINE COMPARTMENT

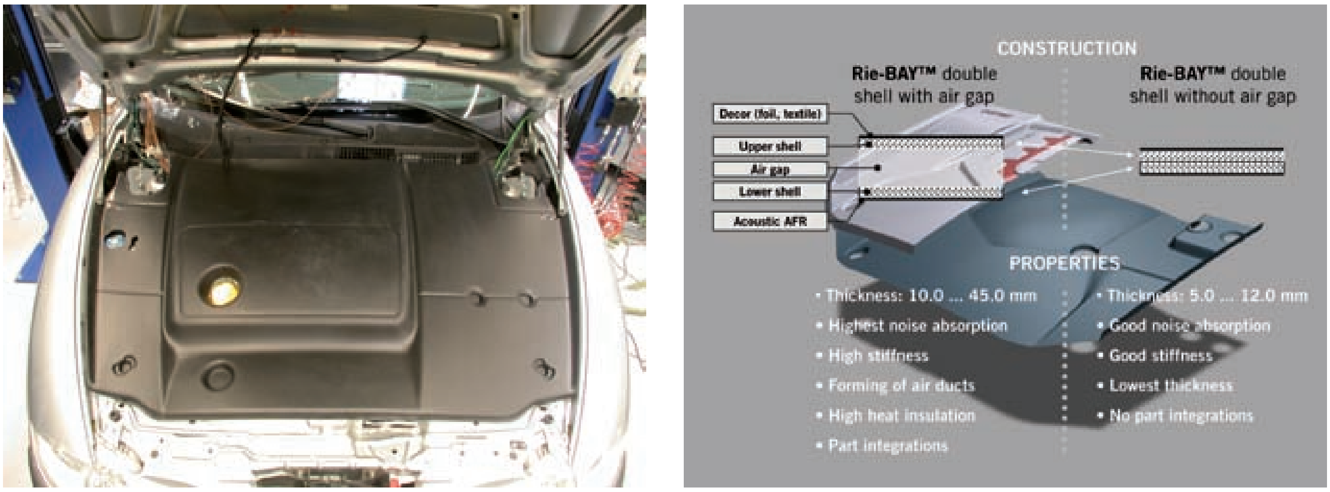
For engine-mounted parts such as engine covers or timing belt covers, injected PP or

	ENGINE-MOUNTED ENCAPSULATION	COMBINATION OF ENGINE-MOUNTED AND BODY-MOUNTED ENCAPSULATION	BODY-MOUNTED ENCAPSULATION
			
WEIGHT INCREASE TO BASELINE VEHICLE	4.7 – 5.2 kg	7.1 – 7.5 kg	4.2 – 4.7 kg
TEMPERATURE INCREASE TO BASELINE VEHICLE AFTER 12 H COOL-DOWN AFTER WARM-UP	6.6	9.4	6.1
CO_2-REDUCTION DURING NEDC OF 1.9L DIESEL ENGINE	1.7 %	2.7 %	1.9 %
THERMAL SAFETY	–/o	–/o	o
ACOUSTIC	+/++	++	+++

❸ Overview of the thermal storage potential of different approaches to engine encapsulation



❹ CO_2 emissions as a function of the starting temperature in the NEDC



❺ Rie-BAY double-shell demonstration part (left) and construction (right)

PA coupled with felt or PU foam-moulded absorbers are widely used. However, these material combinations are not suitable for a complete encapsulation due to their high weight (between 3 and 4 kg/m²).

Body-mounted acoustic parts are often made of glass wool or cotton felt, with phenolic resin as a binder, covered with treated nonwovens. Besides the good economics, these materials provide overall good flammability resistance. However, the resin has some disadvantages such as odour and VOC (volatile organic compounds) emissions.

In recent years, Rieter has developed acoustic materials with sufficient mechanical resistance to be moulded into self-supporting parts. One example is “KEST” from Rieter, a needled fibre material consisting of a combination of PP and a variable amount of glass fibres in order to balance acoustic and mechanical properties. However, such materials are not suited to the high temperatures close to an engine since they are limited to 140 °C (continuous engine surface temperature). There was therefore a need to develop materials that combine mechanical strength, acoustic effectiveness, heat insulation and heat stability.

The frequency range of maximum acoustic absorption can also be adjusted thanks to the lamination of an acoustic resistive scrim (airflow resistance in the range of 1000 MKS rayls). To prevent fluid absorption into the fibres, to guarantee flammability compliance (Vertical UL94 V0) and to provide acceptable aesthetic appearance, a proper choice of surface layers such as nonwovens or films is important.

ACOUSTIC OPTIMISATION

While lightweight materials often provide the necessary thermal insulation, the optimisation of their acoustic properties requires deeper consideration. If, for example, a high acoustic transmission loss is required for the encapsulation, heavy materials are often used, thus largely cancelling out the CO_2 reduction. Rieter used a REVAMP-SEA model of the engine compartment to examine two different material families. The first material family is based on the acoustic mass-spring concept (solid layer plus foam) while the second is based on the combination of a highly porous layer with low airflow resistance and a compressed porous layer with a high airflow resistance. The latter was developed and patented by Rieter under the brand name “Rieter Ultra Light”.

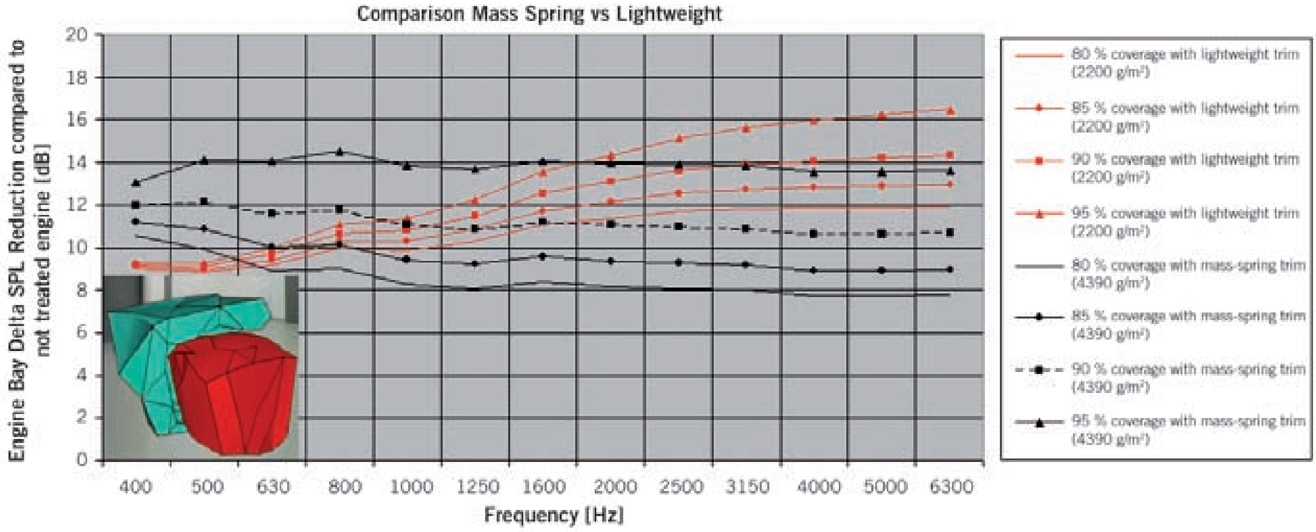
❼ shows that the engine-mounted encapsulation with the lightweight absorbing-insulating material provides a greater noise reduction to the engine compartment than the heavier mass-spring solution, based on a realistic area coverage of 80 %.

The lightweight thermo-acoustic materials are easily mouldable for producing self-supporting parts with a stable shape and stable mechanical properties. At the

REFERENCE	CHARACTERISTICS	B-SEGMENT	C-/D-SEGMENT	E-SEGMENT
STANDARD SYSTEM TODAY	Material Engine/ Engine Bay Covers	PP-injection moulding	PP- or PA-injection moulding	PP- or PA-injection moulding (bi-color!)
	Number of parts	6	5 … 6	8
	Weight (kg)	1.5	2.2	6.3 incl. air inlet
RIE-BAY-PANEL	Material Rie-BAY™-Panel	HT-structural fibre	HT-structural fibre	HT-structural fibre + sealing parts + air inlet
	Number of parts	1	1	1
	Weight (kg)	0.7 (–53 %)	1.5 … 1.8 –21 % … 33 %	4.1 (–36 %)
	Cost reduction potential	+++	+ … ++	+

Rem.: Integration of sealing function (replacement EPDM-seals) not included in B- and D-segment.

❻ Cost and weight comparison for four vehicle segments



❼ Engine compartment sound pressure level reduction due to engine-mounted encapsulation as a function of the area coverage

same time, they are thick enough to guarantee acoustic and thermal insulation.

The local wall thickness and density must be adjustable to comply with the available space and, according to the characteristics of the emitting noise sources, must provide a balance between acoustic absorption, insulation and heat insulation.

The area weight of the entire concept, which integrates the acoustic and thermal insulation functions, is less than 2.5 kg/m². Therefore, they allow the design of self-supporting parts that are able to cover large surfaces.

CONCLUSIONS

It is foreseeable that the internal combustion engines of future vehicles will require closed encapsulations in order to improve fuel efficiency and reduce exterior noise. This poses tremendous challenges for car manufacturers in terms of added complexity, weight, cost and thermal safety problems.

The adoption of innovative system and component architectures, as shown by the Rie-BAY panel from Rieter, can achieve almost complete encapsulation. The CO_2 reduction of 2.5 g/km is significant. What is more, advanced internal combustion engine concepts will be perceived by the end customer as being quieter.

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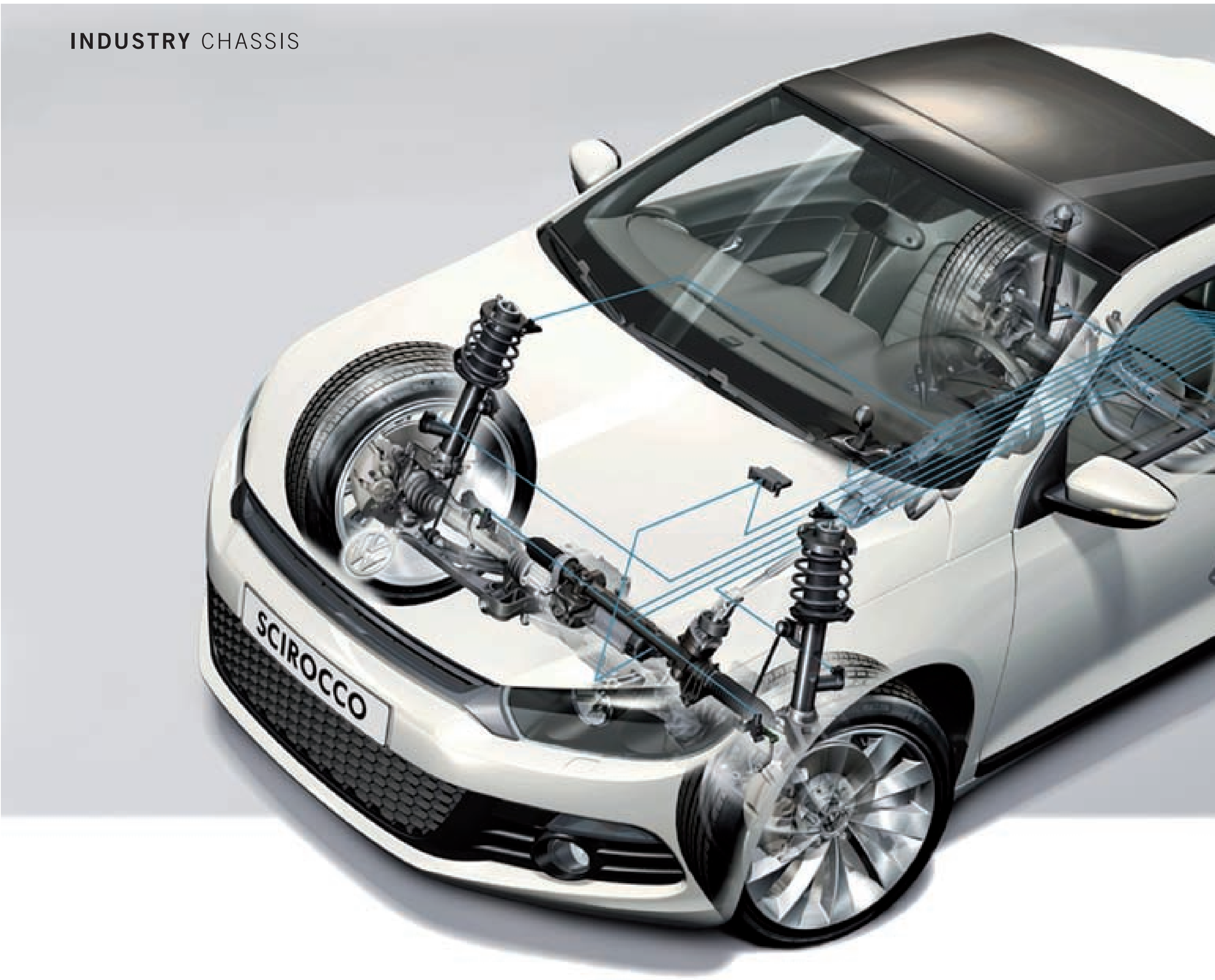

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INTRODUCTION

The level of OEM input in developing control-unit-based systems ranges from adapting externally generated standard software algorithms without the supplier disclosing important system properties ("black box") to developing algorithms in-house from start to finish ("white box"). There are many different levels of involvement in between, each depending on the strategic alignment preferred for the specific project and/or available (staff) capacity.

APPLICATION: ADAPTIVE CHASSIS CONTROL

The adaptive chassis control from Volkswagen uses electrically adjustable dampers and steering that largely couple the qualities of sporty stiff design and driving comfort, ❶. The driver has the option of choosing between three presets – Normal, Sports and Comfort – at the press of a button. A control algorithm constantly matches the shock-absorbing forces to suit road type and driving situation. Information on the position and dynamics of the wheels is delivered from three displacement sensors; three body acceleration sensors determine body movement; CAN signals from various electrical vehicle systems provide data on longitudinal and transverse dynamics.

FUNCTION SOFTWARE DEVELOPMENT

Development, ❷, took place in conformity with the V-model process. Production readiness was achieved in iteration cycles (A-prototype to mass production) in line

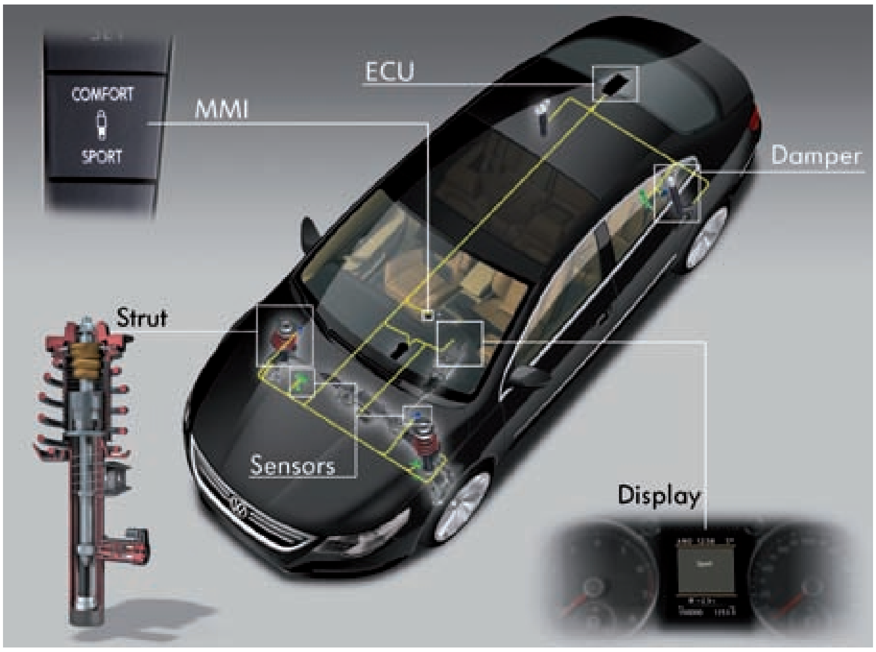
MODEL-BASED ALGORITHM DEVELOPMENT

AUTOMATED FROM THE IDEA TO THE PRODUCTION CONTROL UNIT

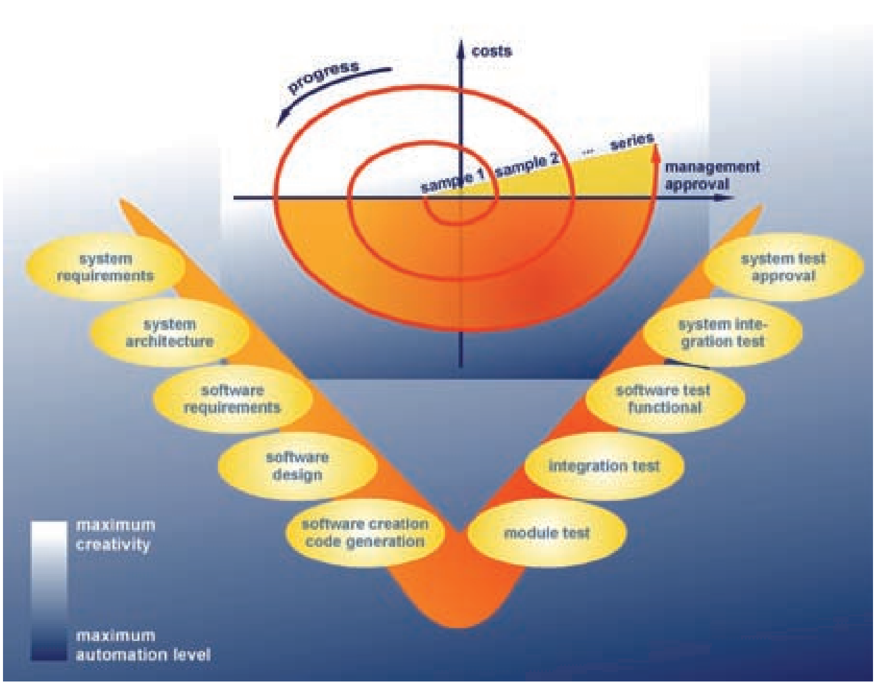
Increasingly shorter development times are demanding an all-embracing algorithm development process covering every stage from rapid prototyping to start of production for electronics of chassis. Under these boundary conditions, possessing full understanding of the system with appropriately structured and protected expertise must also be the aim of the OEM. This is why Volkswagen realized the core elements of the adaptive chassis control in a project conducted jointly with IAV GmbH.

with the spiral model. A central aim of developing function software in-house is to transfer the expertise of Volkswagen's chassis developers to a product with maximum added value for the consumer. This knowledge can be found in the upper and middle section of the left-hand branch in ②. The tip and right-hand branch describe mechanisms for validating the system and serve to meet the requirements that are placed on quality. This scope of activities is extensively automated in a tool chain, making it possible to run through development and validation loops in a highly time-efficient manner.

Function software development is broken down into four principal components: requirements analysis, modeling, tool chain (= realization) and strategies for testing. The last three points are elucidated in greater detail in the sections that follow, after treating the software architecture.



❶ Overview of the system components for the adaptive chassis control at Volkswagen



❷ Development cycles in the V model

SOFTWARE ARCHITECTURE

The software architecture selected is based on a clear division between the basic software developed by the control-unit supplier and the in-house function software, ❸. Both blocks are interlinked by means of an interface and extensively modularized. By way of example here, ③ shows the calibration interface in the basic software. This way, OEM and control-unit supplier are each left to concentrate on their core competencies. Individual modules can be encapsulated, facilitating exchangeability or providing the capability of making changes.

Here, the functional software is solely the responsibility of the OEM. This means the OEM is responsible for defining the concept for the necessary release tests as well as for conducting them. Necessary supplements are implemented in the tool chain and the model for this purpose.

The basic software provides the infrastructure on a control unit. Volkswagen also uses a hardware independent standard software core which, among other functions, defines the various interfaces, for example to the communication layer (CAN, TP etc.), to the task manager (Osek operating system) or to the control-unit hardware (for example hardware abstraction layer for digital I/O, analog I/O, PWM). The algorithm development process described here assumes that the basic software, with is respective components, is given, and is not elucidated in further detail.

MODELING

The function software is modeled graphically using Matlab/Simulink software. This ensures a high degree of system transparency both for the algorithm developers as well as for the internal users. Modeling is covered by parts of the left-hand branch of the V-model from ②.

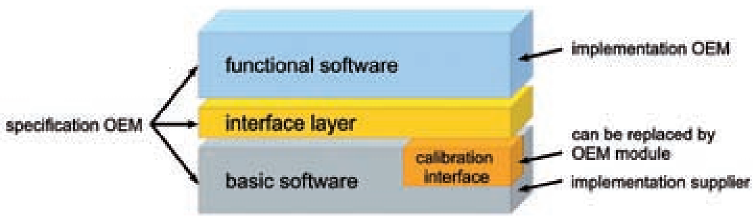
Wherever possible, the subsequent target vehicle is immediately selected as the test environment. Directly after modeling, that means after programming the function requirements the control unit is updated and an initial verification carried out together with staff from the algorithm-development and test-drive teams. In terms of time efficiency, transfer of internal expertise and avoiding interface problems, this somewhat unconventional approach has proven to be highly successful.

The model itself is hierarchically structured and contains functional subsystems at the lowest level, ❹. These are managed in a library structure. This ensures straightforward file versioning and the capability of tracing changes. The function software uses floating-point arithmetic almost exclusively. In view of the many filters used, this is particularly recommended. Conformity between function model and implementation model is a further advantage.

Emphasis must be given to the use of several time slices from 1-ms to 1-s intervals. Consequently, the various subsystems run at different “speeds” in accordance with the specific requirements placed on their performance. Here, a time slice is allocated to each subsystem. The interface concept ensures the desired data consistency between the subsystems.

All parameters and signals defined for calibration as well as for the interfaces to the basic software are filed as Asap variables. This facilitates further processing through the subsequent tool chain and automatically guarantees consistency of data. Furthermore, the recognition effect is extremely high for signals and parameters as the measurement and calibration software uses both the same functional breakdown as well as the same nomenclature.

When changes are made to the model – function description excluded – the documentation is updated automatically. Func-



❸ Overview of the components of the software architecture tion programming takes place on the basis of standardized guidelines (style guides). This generates a largely uniform style of programming void of individual preferences. A rule checker automatically checks for correct implementation and indicates where changes need to be made.

Automated script processes furthermore support programming activities prone to error, such as in relation to nomenclature consistency. The aids presented satisfy a central aspect of quality assurance as early on as the model level.

TOOL CHAIN

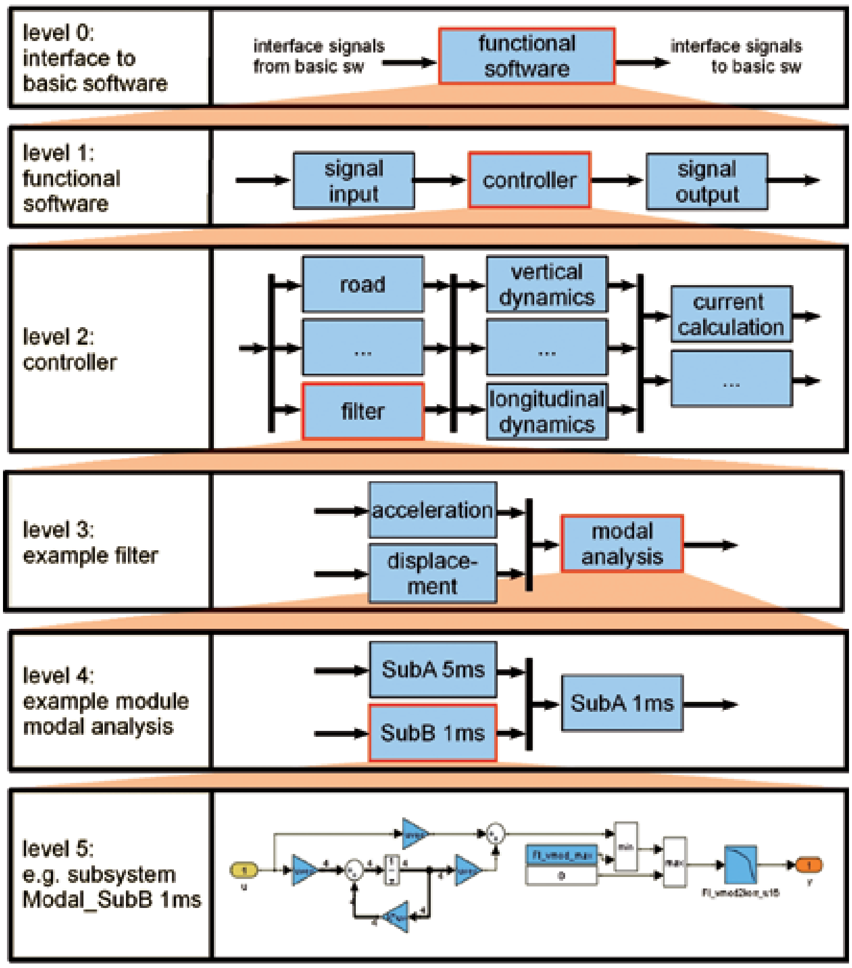
The tool chain sequence is presented in ❺. In this application, the tool chain covers the key parts of code generation as well as module and integration testing at the laboratory workstation or on the HiL. A further defining property of the concept is the ability to use it on a start-to-finish basis, that means from the early rapid-prototyping phase to the start of system production.

The tool chain's input variable is the graphic Matlab/Simulink model with the Asap parameters and signals from the workspace. Both the rule checker for verifying adherence to the modeling guidelines as well as the Docu-Gen module for generating the documentation were already used during the modeling process and are employed here once again.

The Code-Gen module generates the c- and h-source files through the real-time workshop in conjunction with tlc scripts. The Calibration-Gen module extracts the Asap variables relevant to calibration and provides them in c-, h- and a2l-files. The Interface-Gen module contains the c-/h-file according to the interface defined between algorithm and basic software. The Object-Gen module generates the obj-file from the existing c- and h-files.

Obj file, original sources for the interface and the a2l-file form the content that is transferred to the component supplier responsible for the basic software. Accordingly, the a2l- and obj-file from the basic software are transferred from there to the algorithm developers. The applicable settings are clearly defined between both parties for operations of the same type.

The A2L-Gen module brings together the relevant a2-files of basic and function software and provides input files for the measurement and calibration software.



❹ Hierarchical structure of the model levels 0 to 5

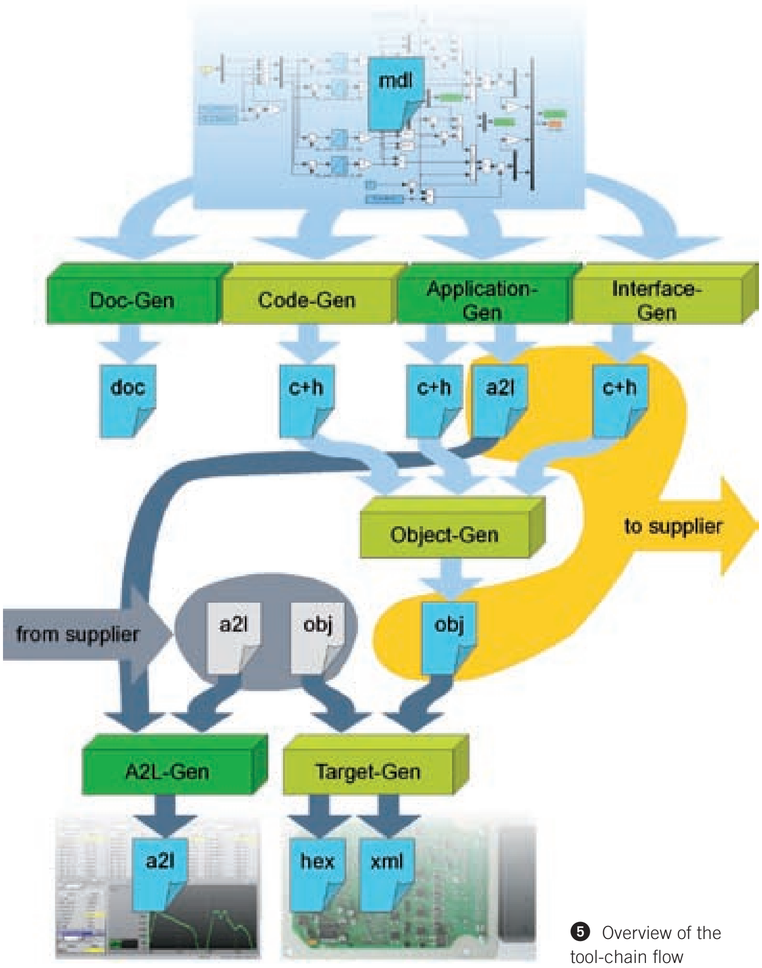
The Target-Gen module generates the control-unit files from the relevant objects of the basic and function software.

Software switches implemented in the tool chain provide the capability of adapting the executable control-unit codes to different calibration interfaces, such as CCP, XCPonCan, XCponEthernet and ETK. An additional a2l file also containing all other signals present in the module on top of the signals defined for the calibration environment can be generated automatically for tests or faults analyses of a more detailed nature.

STRATEGIES FOR TESTING FUNCTION SOFTWARE DEVELOPMENT ACTIVITIES

The testing procedures defined for the production release process range from deep code level via the integration and the interaction of various function blocks to real-life driving maneuvers. The following criteria are worthy of particular attention:

- To a very large extent, the tests can be conducted independently of the basic software from the control-unit supplier.
- As a result, test scheduling is detached from any availability of the software developed externally.
- Very good test coverage is ensured at all levels.
- The test procedures running outside the vehicle are extensively automated and designed for high reproducibility and time efficiency.
- Manual checks only take place for resource-related needs (memory, runtime) and errata.
- Testing is conducted on production hardware, thus providing high informative value.
- Altogether, these attributes produce a high quality of testing and thus high system quality.



❺ Overview of the tool-chain flow

Code testing is broken down into module, integration and system tests. The module and integration tests essentially check control unit behavior against model functionality. The quality of code coming from the generator is examined and assessed in reviews that check the C source against the Simulink model as the status of model development progresses. In this context, autocode generation has been shown to achieve very good results in relation to memory, runtime and readability.

The function tests consist of module and system tests. The module tests are automated and take place on the HiL whereas the majority of system tests are performed in the vehicle.

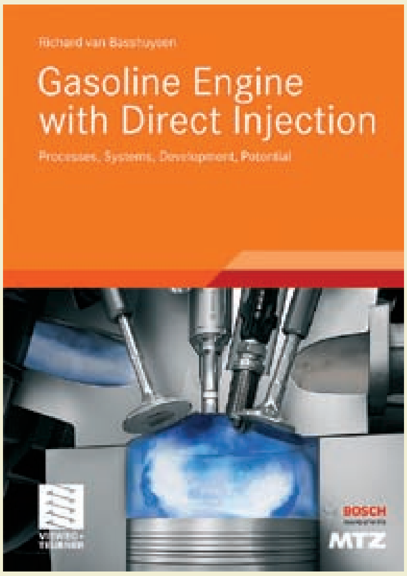
SUMMARY AND OUTLOOK

The approach, outlined by Volkswagen and IAV, provides the following benefits for the development of an adaptive chassis control:

: In terms of function, the system can be tailored in-house to suit the specific characteristic vehicle properties and requirements exactly and without compromise.
: Stringent modeling instructions generate standardization effects and make the model easy to understand and maintain. The function modules can be exchanged, providing easy adaptability.
: Functional changes can be implemented very quickly; it takes approximately 10 min to run through the tool chain from feeding in the model to the availability of the control-unit code and the subsequent process of writing it into the control-unit flash memory.
: Fully automated code generation and code integration saves a considerable amount of time. The method also avoids errors, producing results of a high quality.
: The flexibility of the model and the tool chain means they can be and are being used without a problem in many derivatives and also at cross-brand level within the group.
: The modularization of software development activities with strict separation of basic and function block allows algorithm development to take place independently of the supplier's basic system. To do this, it is necessary to define all interfaces clearly and in full.
: As external development partners only receive objects, expertise underlying the model is protected as far as possible.
: The fields of activity are clearly divided between OEM and external development partners. All parties retain their respective sovereignty within their specialist domain.

The in-house-development activity presented has been in use in several Volkswagen Group car derivatives and brands since the end of 2007. Synergy effects are opening up for follow-on projects, just as they are for parallel algorithm development activities in the chassis and suspension segment. Adaptation or conversion to standards, such as Autosar, is not a problem with the approach described in this article.

Gasoline Engines are the answer to the challenges of future



Richard van Basshuysen

Gasoline Engine with Direct Injection

Processes, Systems, Development, Potential

2009. xviii, 437 pp. With 399 Fig. Hardc. EUR 49,00
ISBN 978-3-8348-0670-3

Direct injection spark-ignition engines are becoming increasingly important, and their potential is still to be fully exploited. Increased power and torque coupled with further reductions in fuel consumption and emissions will be the clear trend for future developments. From today's perspective, the key technologies driving this development will be new fuel injection and combustion processes. The book presents the latest developments, illustrates and evaluates engine concepts such as downsizing and describes the requirements that have to be met by materials and operating fluids. The outlook at the end of the book discusses whether future spark-ignition engines will achieve the same level as diesel engines.

authors | editors

Dr.-Ing. E. h. Richard van Basshuysen was Head of Development for premium class vehicles and for engine and transmission development at Audi. Today, he is editor of the magazines ATZ and MTZ. The editor was supported by a distinguished team of authors consisting of 22 experts and scientists from industry and universities.

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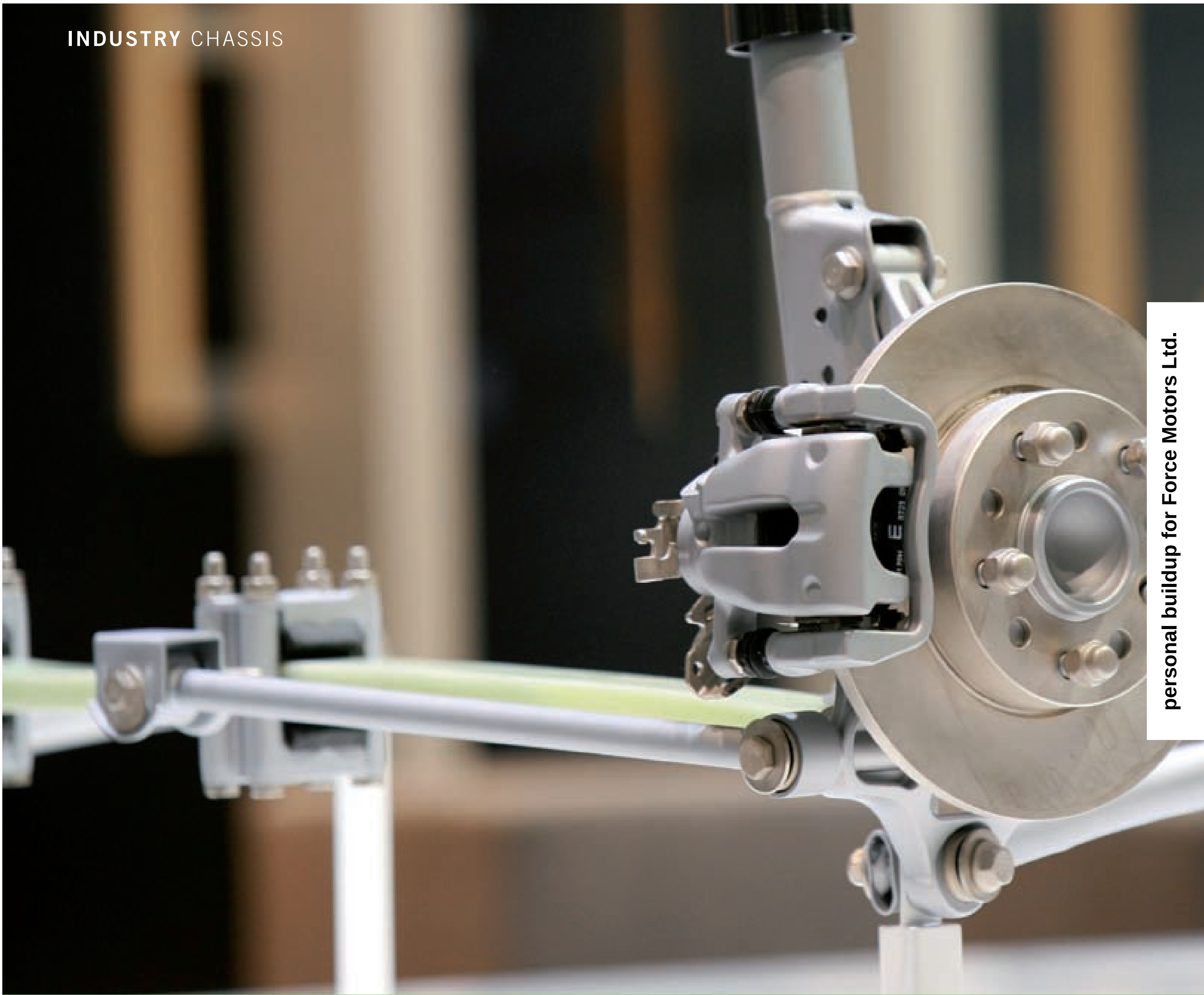

PISTON ROD VIBRATIONS IN DAMPER MODULES – CAUSES AND REMEDIES

Different stiffness values of dampers and top mounts, which connect the damper with the body or a subframe can cause strong piston rod vibrations. In consequence this may contribute to bad NVH properties of damper modules and can cause disturbing noise in the vehicle. Research at ZF indicated that changing the shape of the damper characteristics or adapting the top mount stiffness to the damping force characteristic are solutions that enable to reduce comfort impairing damper vibrations.

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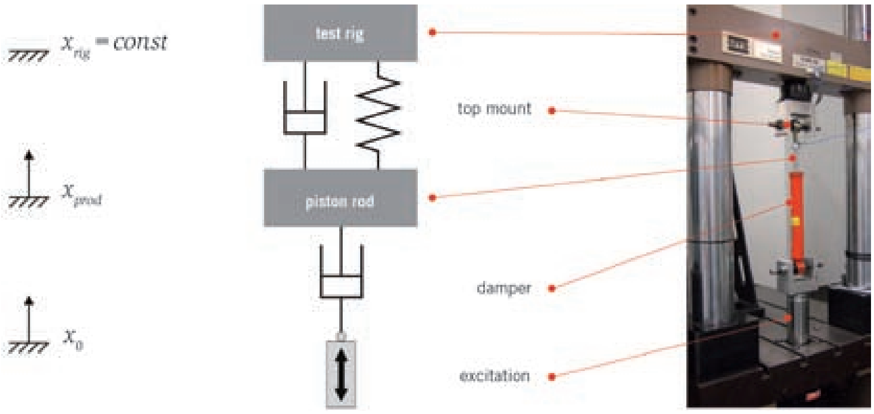
VIRTUAL METHODS

Hydraulic shock absorbers featuring usually digressive valve characteristics have a dynamic stiffness that increases with rising excitation frequency. This feature affects the NVH properties of damper modules and – due to the vibration transmission by the body attachment – that of the entire vehicle in some cases. The dynamic stiffness leads to longer, less damped relaxation phases of suspension vibrations and, in interaction with other vibrating chassis components, can cause secondary resonances to occur, which lead to noticeable and, in some cases, audible impacts on ride performance [1]. Dampers are therefore normally mounted via rubber-metal parts to the vehicle body or a subframe. These mounts help to reduce the dynamic stiffness of the damper module. A high level of damper module stiffness leads to high body accelerations, which are assessed uncomfortable [2]. Within the following text, the term "damper module" is used solely for the subsystem of damper plus upper mount, the so called top mount.

The force between the actual hydraulic damping element and the top mount is supported by an inertia element, the piston rod mass. For this reason, special diligence is also required when designing the mount stiffness, as otherwise the inertia element can be excited to undesired vibrations. In this article, ideas concerning the modeling of the subsystem of shock absorber and top mount are developed that describe the vibration behavior of the piston rod. In particular, the contributions of hydraulic characteristic curves and damper elasticity to the piston rod vibrations are explained. Based on these ideas, it will be examined how different top mount stiffnesses can contribute to the optimization of this vibration behavior.

Usually, NVH investigations and optimizations during the chassis development are largely carried out using experimental methods. Sensible virtual additions to the methods mostly refer to either describing/deriving hydraulic damper characteristic curves as F-v elements or to MBS investigations of corner modules or entire vehicles in the low frequency range up to approximately 30 Hz. In the early stages of vehicle development, there are many degrees of freedom for changing parameters, and therefore investigations of subsystems on test rigs are usual. In contrary, during later stages of development, only optimizations of details are possible which are validated with vehicle measurements, for instance impact measurements on a 4-post test rig. However, vehicle fine tuning does usually apply not earlier than in this phase, since body parameters are frequently not finally defined earlier on. Moreover, the hydraulic damper characteristics and vibration parameters of the damper mounts can be varied more easily than other properties and at a later point in time. Therefore, the tuning of them is very effective to the vehicle dynamics performance.

Experience shows that finding an optimal balance between handling and ride targets is not always easy. Ride comfort is a combination of acoustic and mechanical vibrations. The methods used at ZF Sachs to optimize vehicle acoustics, shown in ATZ 5/2009 [6], are enhanced through the knowledge of this contribution and are enriched through a new approach. The procedure differs from others in the aspect that fundamental knowledge is gained using physical models limited to sub-aspects, but without imposing the requirements of technical-economic practicability.



❶ Test rig with damper module

THE VIBRATION ENGINEERING DESIGN OF VEHICLE DAMPER MODULES

Within the context of this work, the simulation is restricted to certain dynamic test rigs, which are often used to evaluate vibration dampers. ❶ shows the schematic design of such a test rig. Damping forces are generated between the piston rod and cylinder, which are transferred via the corresponding mounts to the vehicle and, in this case, to the test rig. For this purpose, the damper is described either using a simple force-velocity characteristic (F-v model) or by means of the pressure-flow calculation (p-Q model) [3]. The main difference between the two is the consideration of both oil elasticities and cylinder wall stiffness in the p-Q model, which leads to a significantly better recording of the dynamic effects.

With respect to the damping medium, the damper represents a closed system, where the oil volume displaced by the relative movement of cylinder and piston rod leads to a pressure change in a gas reservoir. The resulting “spring part” of the damping force is taken into account in the p-Q model, however, it plays a subordinate role in the following considerations. The focus of these investigations lies on the properties and design of the top mount. At the same time, it is assumed that the rubber-metal top mount is mainly acting as a spring. So, the “damping forces” parts of the top-mount can be neglected. From the arrangement of the piston rod in ① it can be seen that non-linearities in the damping characteristics and/or in the top mount characteristics can cause the piston rod to vibrate. Experience shows that these high frequency vibrations pass through the top mount into the vehicle body and are perceived as disturbing noise by the passengers. Therefore, the goal of a good damper module design should be to avoid piston rod vibrations, among others.

BALANCE OF THE PISTON ROD FORCES

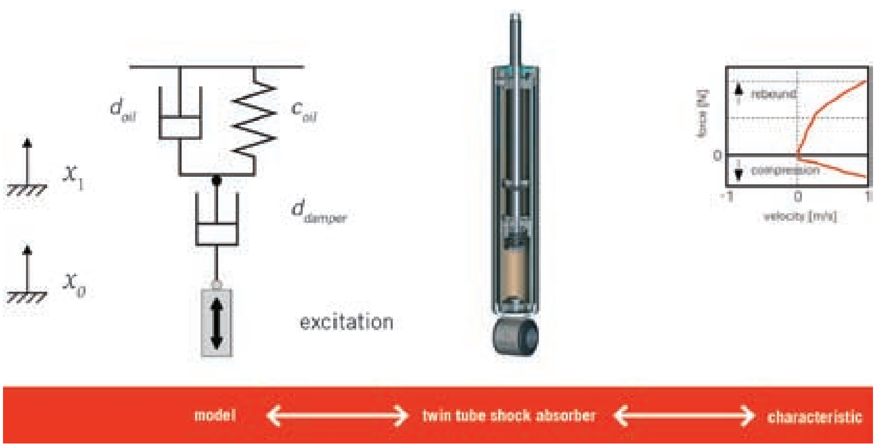
① shows the piston rod force equilibrium:

EQ. 1	$\ddot{x}_{prod} = (F_{Damper} + F_{top\ mount})/m_{prod}$

In Eq. 1 $\ddot{x}_{prod}$ is the piston rod acceleration, F_{Damper} the damping force, $F_{top\ mount}$ the top mount force, and m_{prod} the mass of the piston rod. The damping force and the top mount force transferred to the vehicle body are not identical. The degree of freedom of the inert piston rod mass must be taken into account: vibrations of the piston rod mass may reach the vehicle body. In technical reality, however, the damper shown in ① as the ideal damper element has elasticities due to the oil compressibility and cylinder wall stiffness.

In [4], therefore, an equivalent schematic for the real damper has been derived that takes these factors into account, ❷. As [4] shows, not all the oil displaced by the piston rod is available for energy dissipation, but instead a portion of it fills certain parasitic energy accumulators, which are formed by these elasticities. The discharging of the parasitic energy accumulators intensifies existing piston rod vibrations when the damper travel changes from rebound to compression mode [4].

The mechanism for creating piston rod vibrations has been shown in [5]. In automotive applications, the form of the F-v characteristic is typically significantly nonlinear, but at the same time, the deflection of the top mount mostly lies in the linear range. As a result, the stiffness ratio between damper and top mount changes where the damping force charac-



❷ Equivalent damper model with oil elasticity teristic is nonlinear. According to Eq. 1, this change forces the mass of the piston rod to a change in motion, leading to the excitation of piston rod vibrations.

PISTON ROD VIBRATIONS IN THE IDEALIZED SYSTEM

The piston rods equation of motion, based on the force equilibrium Eq. 1, highlights this relationship, when one considers zero damper elasticities ($x_1 = x_{prod}$) and zero damping of the top mount ($d_{tm} = 0$):

EQ. 2

$$\ddot{x}_{prod} + \frac{d_D}{m_{prod}}\dot{x}_{prod} + \frac{c_{tm}}{m_{prod}}x_{prod} = \frac{d_D}{m_{prod}}\dot{x}_0$$

Here, the piston rod speed $\dot{x}_{prod}$ is connected to the damper speed v_1 through the relationship:

EQ. 3

$$\dot{x}_{prod} = \dot{x}_0 - v_1$$

If one defines the stiffness ratio t_T [5], which has the dimension of time, as:

EQ. 4

$$t_T = \frac{d_D}{c_{tm}}$$

then Eq. 2 can also be written as follows:

EQ. 5

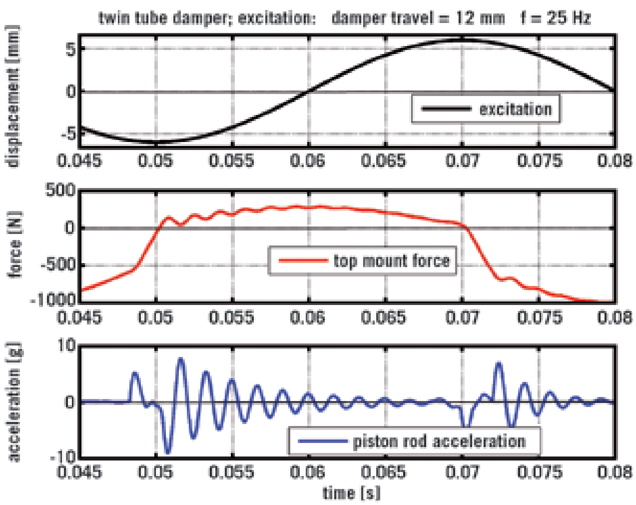
$$\ddot{x}_{prod} = \omega^2(t_T(\dot{x}_0 - \dot{x}_{prod}) - x_{prod})$$

Here, ω is the Eigenfrequency of the piston rod. The inner bracket term on the right-hand side of Eq. 5 is the damper velocity. Assuming the damper velocity changes uniformly, the piston rod moves relative to the top mount with a specific uniform acceleration. If, during this movement, the stiffness ratio t_T changes at time t because the damper's F-v characteristic is nonlinear, the change in the previously constant acceleration of the piston rod must be correspondingly strong.

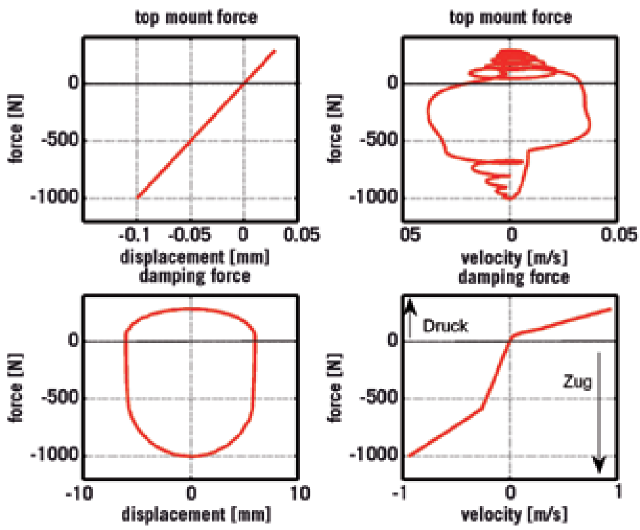
❸ is designed to illustrate this result, in which negative forces from here on mean rebound and positive forces compression. A system according to ① with a F-v model for the damper not taking the damper elasticities and damping of the top mount into account, has been simulated at a sinusoidal excitation of 25 Hz and 6 mm amplitude. The excitation of 25 Hz and 6 mm amplitude is selected because, in this case, considerable piston rod vibrations occur. The model has an elementary mechanical structure and the model equations are not subject to any restrictions. Consequently, the piston rod motion is correctly determined for this idealized system at any excitations. However, fluid flow effects of the valve system, as occurring in real dampers, are not taken into account in this model.

Information about the magnitude of the piston rod vibrations can also be derived from Eq. 5. The term $\omega^2 x_{prod}$ is the acceleration of the piston rod, caused by the top mount stiffness. The acceleration is proportional to the piston rod travel. The term $\omega^2(t_T(\dot{x}_0 - \dot{x}_{prod}))$ corresponds to the acceleration, which is determined by the excitation and the damper stiffness. This term obviously depends on the stiffness ratio t_T and the acceleration excited. At equal damper speed and for a constant stiffness ratio, the amplitudes of the piston rod vibrations are therefore proportional to the acceleration excited.

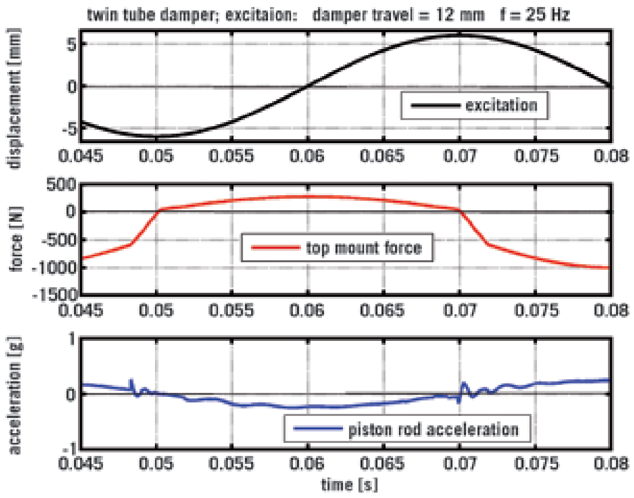
The dynamic properties of such a model are shown in ❹. Noteworthy are the two non-linearities in the damping force characteristic, which together with the linear top mount characteristic, result in the typical vibrations of the piston rod which can be observed in corresponding rig tests [5]. The hypothesis from [5], that a significant reason for the occurrence of piston rod vibrations in damper modules is due to a change in stiffness ratio between damper and top mount at the points where the damping force characteristic is nonlinear, is confirmed by the fact that such a simple model reproduces these vibrations. In [5], a procedure was specified for creating a top mount characteristic adjusted to the damping force characteristic, thereby largely eliminating piston rod vibrations. If this procedure, which is called stiffness adaptation, is also applied to the model used above, a large reduction in piston rod vibrations should be observed. The result is shown in ❺. It should be noted that the scaling of the piston rod acceleration has been amended by a factor of ten compared to ③.



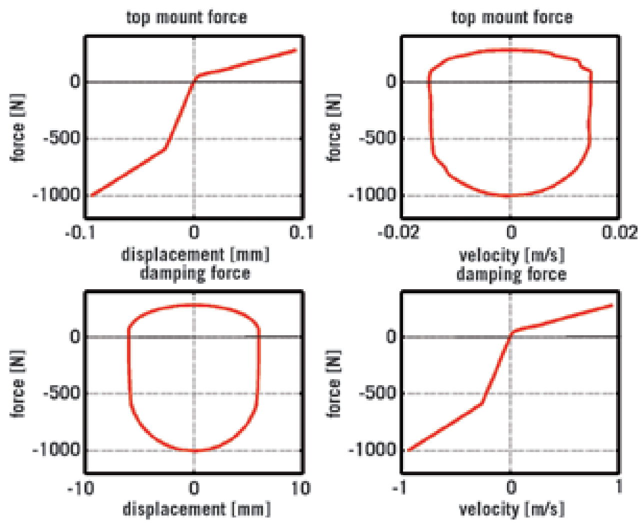
❸ Excitation, top mount force and piston rod acceleration



❹ Dynamic force characteristic of a damper without elasticity and a top mount without damping



❺ Excitation, top mount force and piston rod acceleration of a stiffness adapted damper module



❻ Dynamic force characteristic of a damper without elasticity and a stiffness adapted top mount without damping

The main result can be seen in the piston rod acceleration, which is similar to a sinus function, with somewhat larger changes only occurring in the area of the damping force stiffness change. The value of the acceleration barely exceeds 0.3g and thus is almost two orders of magnitude smaller than in the previous example with conventional top mount stiffness. The run of the damping force curve, middle curve in ⑤, shows no disruptive irregularities, which is predetermined by the damping force characteristic, bottom-right curve in ❻.The representations of the force dependencies in ⑥ show the idea of adaptive module stiffness. The damper's F-v characteristic, ⑥ bottom-right, corresponds in qualitative terms to the F-x characteristic of the top mount, ⑥ top-left. The stiffness ratio of top mount versus damper is constant over the entire deflection range, namely it is the stiffness ratio t_T determined by Eq. 4. This demonstrates that in an idealized system with concentrated parameters, piston rod vibrations are excited by changing stiffness ratios.

SENSITIVITY CONSIDERATIONS IN STIFFNESS ADAPTATION

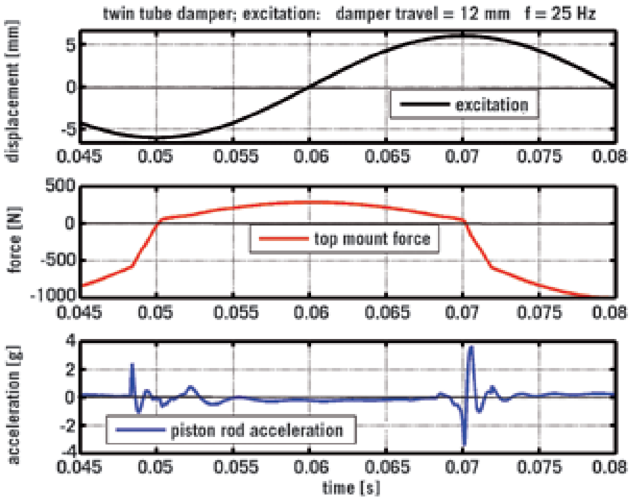
In reality, the general term "stiffness" typically comprises both spring and damping ratios. For example, a real damper also exerts spring forces and a real top mount also generates damping force contributions. This fact is expressed in complex maths in a phase angle δ_{tm} between force and displacement in the top mount and δ_D between force and speed in the damper. If one inserts complex dimensions for the displacements and forces, this produces the complex stiffness ratio:

EQ. 6
$$\frac{1}{t_T^*} = \frac{i\omega\hat{x}_D}{\hat{x}_{prod}} e^{i(\delta_{Kl}-\delta_D)}$$

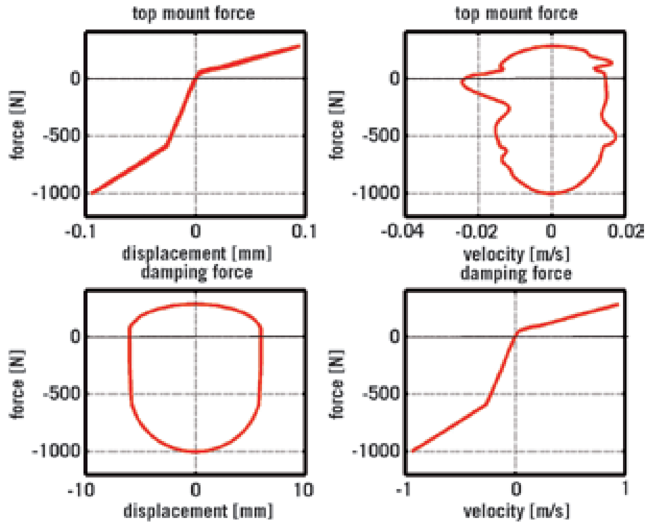
The factors marked "^" are the maximum values. For a pure damper or a pure spring, the phase angles disappear and Eq. 6 is transferred to Eq. 4.

The adjustment of the dynamic characteristic properties of damper and top mount will be critically important for the success of the method. If this adjustment is only unsatisfactorily successful, changes in the stiffness ratio and thus higher values for the piston rod acceleration are inevitable. Firstly, a model with an idealized damper (F-v characteristic without damper elasticities) and a stiffness-adapted top mount with little damping are considered. As shown in ❼, the maximum piston rod accelerations immediately increase to values of around 4g. The piston rod force itself appears only a bit different than in ⑤. The cause of this acceleration increase lies in fact that the adaptation of the force characteristic is no longer complete. The top mount's damping forces have not been considered in the damping force characteristic. This can be seen in ❽. The damping force in the top mount leads to a small hysteresis in the force-displacement characteristic, ⑧ top left, and thus to a small shift of the non-linearities. As a result, the characteristic curve adaptation remains incomplete and increased piston rod accelerations are the result.

If one models the damper now taking the elasticities of oil and cylinder wall into account [3], the stiffness adaptation performed with the idealized model is further violated and even greater acceleration values are expected. Results of corresponding simulations are shown in [5]. The comparison to a non-adapted damper module is also shown in [5]. The acceleration values resulting from this are around twice as large as the ones shown in ③. In this context, it must be noted that, even with a linear F-v damping characteristic, the system-immanent elasticities of oil and cylinder wall add to increased piston rod vibrations, as shown in ❾. The elasticity reservoirs, which are filled differently in rebound and compression, lead to a damping force hysteresis that differs noticeably from the spring force hysteresis of the top mount and thus develops into a disruption of the module's stiffness adaptation. This disruption, caused by non-ideally adjusted characteristics, makes the reciprocal setting of damper and mount characteristic more complex.



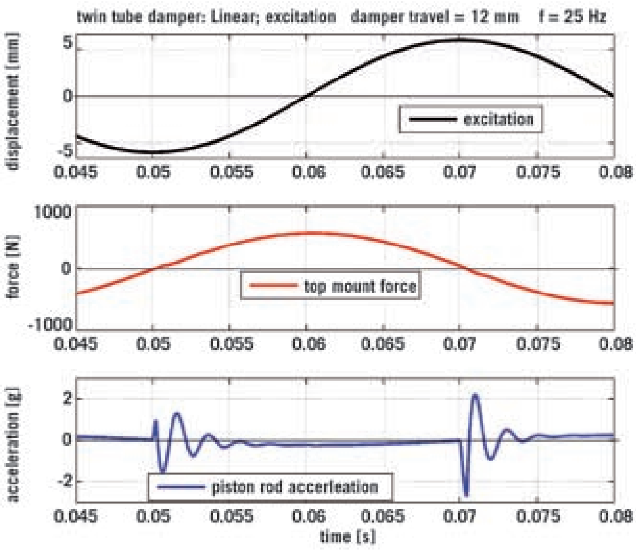
❼ Excitation, top mount force and piston rod acceleration of a stiffness adapted damper module with little top mount damping



❽ Dynamic force characteristic of a damper without elasticity and a stiffness adapted top mount with little damping

SUMMARY AND OUTLOOK

The simulation results described here show the great potential of the so-called stiffness adaptation in damper modules because it involves solving the root cause of piston rod vibrations. Damper and top mount stiffness values that are not adjusted to each other can cause strong piston rod vibrations and this can contribute to disturbing noise in the vehicle. Apart from changing the shape of the damper characteristics [6, 7, 8], the top mount stiffness adaptation to the damping force characteristic appears to be another viable instrument for reducing comfort impairing damper vibrations. In light of typical production tolerances and other influences the stiffness adaptation method's suitability for practical use cannot be conclusively judged at this time. The technical implementation of a top mount with the required nonlinear force characteristic certainly seems possible in principle. This may be achieved by using pre-loaded buffers, applying Euler's strut or the mechanics of honeycomb structures [9]. The investigations planned next will show to what extent tolerances in the dynamic properties of dampers and top mount have a negative impact on the stiffness adaptation.



❾ Excitation, top mount force and piston rod acceleration of a stiffness adapted damper module with little top mount damping, linear F-v-characteristic and including the damper elasticity

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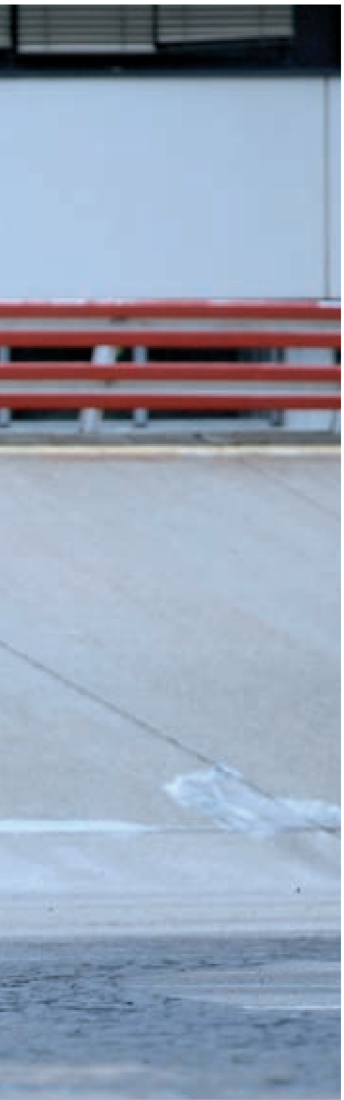
DRIVER-FOCUSED CONFIGURATION OF VANS – NEW APPROACHES, FUTURE DEVELOPMENTS

With the high performance of modern vans, their driving performance also has to keep up with. Therefore the conventional assessment of handling and steering performance will be extended by methods derived from control theory and NVH (Noise, Vibrations and Harshness). This paper takes Mercedes-Benz vans by way of example to present a frequency-orientated process which both enables the vehicle behaviour to be optimised systematically and takes into account the influence exerted by the driver.



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DRIVER-VEHICLE SYSTEM

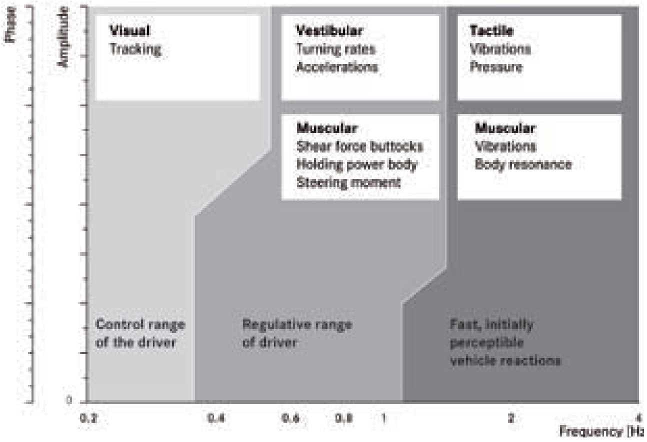
A fundamental objective in a driver optimised vehicle design is to improve the predictability of how the vehicle reacts. For instance, a vehicle in comparable road situations should display comparable driving and steering reactions. The reaction of the vehicle body should be simple and also understandable for drivers with little driving experience. As the actions of the regulating systems play a key role in the steering activity of the driver, both in demanding as well as less demanding driving situations, it is necessary to study the driver-vehicle system from a technical control viewpoint.

However, conventional methods of evaluating the driving and steering behaviour consider the vehicle configuration predominantly from the aspect of time, it is therefore extremely difficult to take a control-centred view. The evaluation criteria mostly describe technical relationships which only display an indirect link to the driver's perception. This means, for example, that the self-steering gradient [8] is frequently taken as the key factor in the driving dynamics to evaluate the subjectively perceived stability and controllability of a vehicle. However, it has been shown that self-steering gradients of about 0.20-0.35 $°s^2/m$ are deemed favourable for the controllability and stability of cars, whereas figures greater than 0.30 $°s^2/m$ are considered favourable for the stability and controllability of heavier commercial vehicles. This means that there is only limited transferability possible between the two vehicle classes. A better approximation of the subjective perception of stability in practice is the stability factor [8] which standardises the self-steering gradients to the wheelbase. However, by not including the seating position in this approximation means that it does not always generate usable correlations with the subjective perception of stability in every vehicle concept.

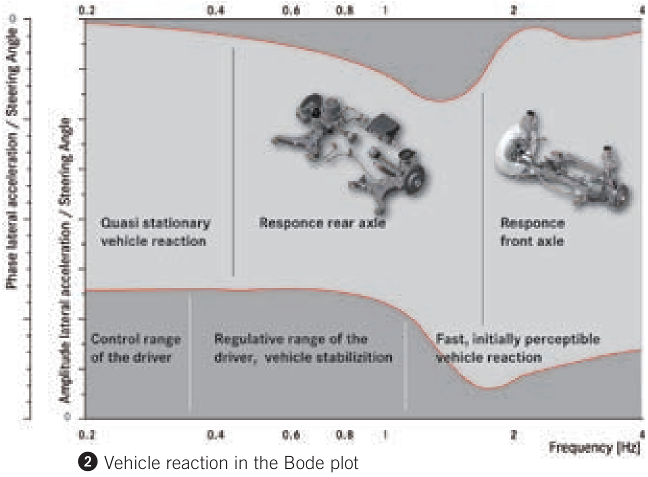
Even so, modern measuring and evaluation methods can be applied to evaluate criteria that are based more around the perception of the driver and less around the technical specifics of the vehicle and as such facilitate improved transferability between vehicle classes and superstructure variants.

DRIVER PERCEPTION IN THE BODE PLOT

As the perception of the driver is highly selective frequency-wise it would be sensible to depict the driver-vehicle reactions in the form of Bode plots . (This plot by Hendrik Wade Bode consists of a graph which logs the magnitude and a phase shift of a complex transfer function [5].) The method of interpreting the Bode plot will now be explained taking as an example the lateral acceleration reaction of a vehicle following a sudden change in the steering angle. Basically, a sudden change in steering angle affects all frequencies of the vehicle simultaneously. However, the driver perceives the individual movement forms of the vehicle reaction sequentially. For example, fast vehicle reactions in the range 1-4 Hz are registered first, ❶. However, they cannot be corrected by the driver because of the unbridgeable time lag in the human capacity to react. They do, however, serve the driver as a predicative instrument for the vehicle reactions that are expected to follow [1, 2, 3]. The body's own information providers are here the muscu-



❶ Driver perception in the Bode diagram



❷ Vehicle reaction in the Bode plot

lar and tactile receptors of the driver and the sensory system of the equilibrium organ. Slower vehicle reactions in the range of approximately 0.3-1 Hz serve the driver as the most important control factor in stabilising the vehicle. The frequency limits vary according to driver and his experience [6]. In this range, control of the vehicle is predominantly effected by the perceptions of the equilibrium organ [4], but also via the muscular receptors, e.g. in the arms, torso and buttocks, and partially also via the driver's visual perception of the environment. The driver can only rely on his visual perception of the environment to be aware of the low frequency act of staying in lane.

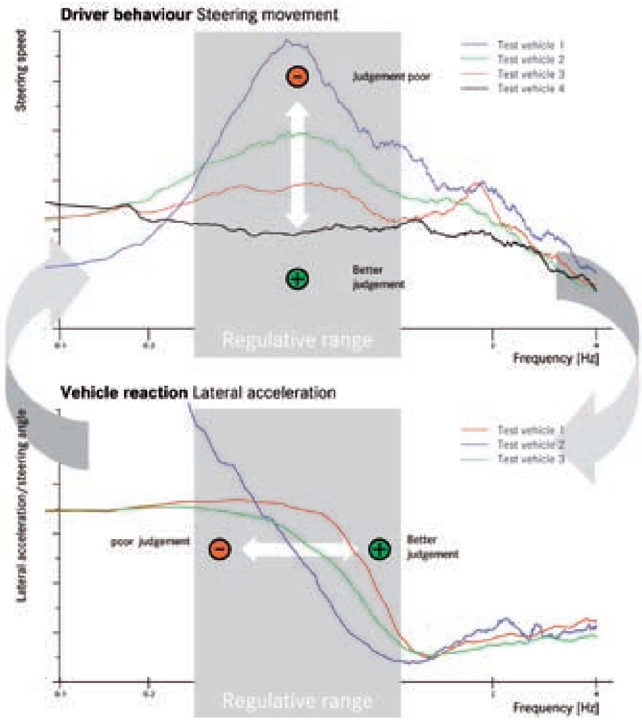
Taking the lateral acceleration reaction of a vehicle caused by steering angle input reveals different and important response areas of the vehicle, ❷. During a step steering input the driver perceives first the quick response of the front axle and then the somewhat slower response of the rear axle.

EXAMPLE – OPTIMISATION OF DIRECTIONAL STABILITY

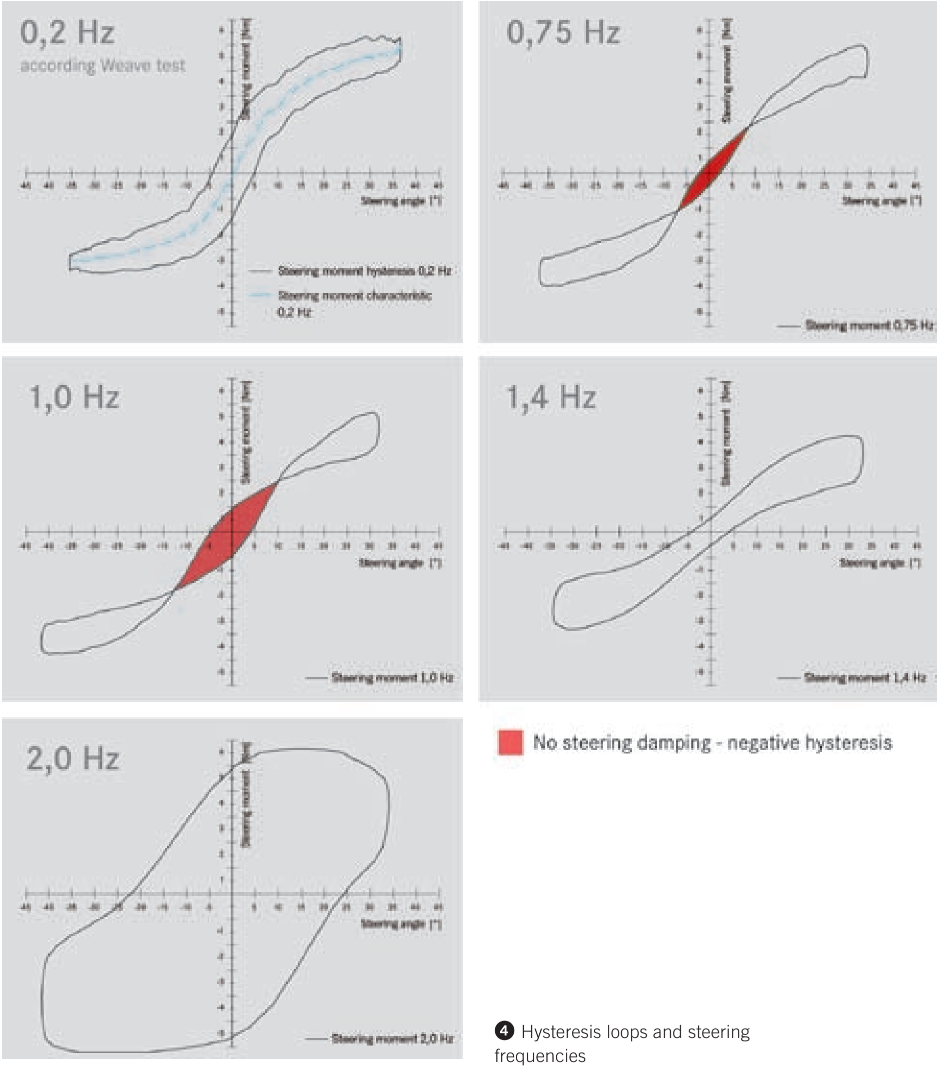
By taking an optimisation of directional stability of a vehicle as an example, we can demonstrate the procedure of vehicle optimisation.

The directional stability of a vehicle is usually evaluated as the degree of deviation of the vehicle from tracking straight and true under interruptive factors such as grooves or side wind. Tests show, however, that the subjective evaluations of directional stability only correlate imprecisely with the actual deviations of the vehicle under interference factors. It has been shown that the control activity of the vehicle constitutes a better criterion to describe insufficient directional stability.

The vehicle reaction perceived from the driver's seat in the form of the lateral acceleration reaction ❸, likewise correlates closely with the evaluations of directional stability. Here it can be seen that a very responsive vehicle with high corner frequency is more positively evaluated than a sluggish vehicle response with low corner frequency. The optimum position of the corner frequency lies in the range of maximum possible control frequencies of the driver or slightly more than this and is thus largely independent of the vehicle configuration. In this configuration the stability of the driver-vehicle control loop is greater than on a lower position of the corner frequency, i.e. a slower response of the vehicle, which is reflected in an overall lower control activity, ③.



❸ Steering movement driver – response behaviour of vehicle

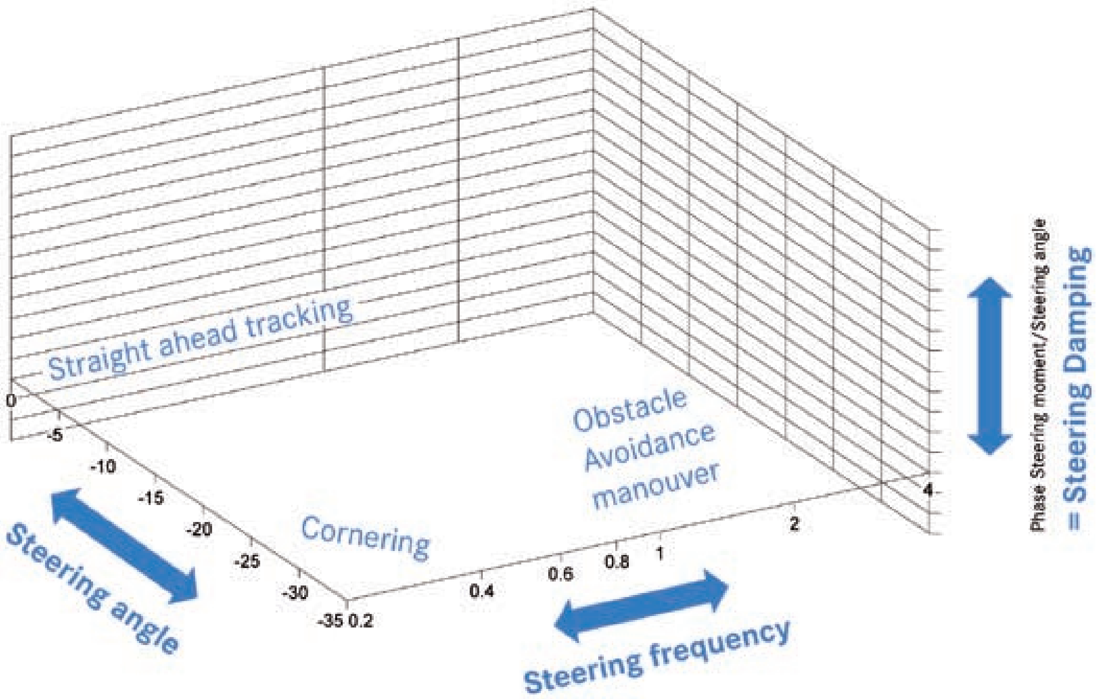


❹ Hysteresis loops and steering frequencies

Although the position of this corner frequency is directly related to the technical criterion of the self-steering gradients, as can be derived from the single-track model [5], it is also dependent of the wheelbase and yaw inertia of the vehicle. This means that a greater and more sluggish commercial vehicle requires a considerably greater self-steering gradient in order to achieve the same position of corner frequency. It has therefore been shown that the optimum value of the technical criterion “self-steering gradient”, depending on vehicle class and superstructure, can be derived from the more general criterion independent of vehicle configuration of “corner frequency of the lateral acceleration reaction” and thus can be derived from the capabilities of human reactions (cf introduction).

It is precisely the seating position of the driver of commercial vehicles, which is well to the fore and above the road surface that also leads to a completely different perception of the steering reaction than of a seating position close to the centre of vehicle gravity such as usually found in cars. This also results in significant differences in the subjective evaluation of the vehicle. However, traditional technical criteria, such as self-steering gradient, stabilisation factor or the yaw are independent of seating position and do not generate evaluation differences. For instance, the directional stability of vans, despite having the same wheelbase and yaw moment, frequently correlate neither with the pure lane deviation under road and side wind influences, nor directly with the stability of the vehicles expressed in the form of self-steering gradients. For example, if insufficient roll stability causes the driver to perceive a lateral acceleration reaction of the vehicle too late, this reduces the subjectively experienced stability of the control loop driver-vehicle and test participants have not only made comments such as “the vehicle feels lax”, but also rather unexpected comments such as “the tracking is worse“ and “the vehicle feels less stable”. Objectively a greater steering activity on the part of the driver can be recorded in these cases.

The influence of the roll movement has already been taken account of in the evaluation criterion “position of the corner frequency” as the lateral acceleration reaction is directly measured at the driver’s seat. The criterion is therefore also a generally applicable evaluation criterion here for directional stability and the stability of the driver-vehicle tandem.



❺ Steering damping – damping map

EXAMPLE – OPTIMISATION OF THE STEERING DAMPING

Another example of applying frequency-oriented methods in evaluating a vehicle is the configuration of the steering damping perceived by the driver. One of the objectives here is to solve the conflict of aims between low steering damping = improved handling, and high steering damping = simpler vehicle control in demanding vehicle situations.

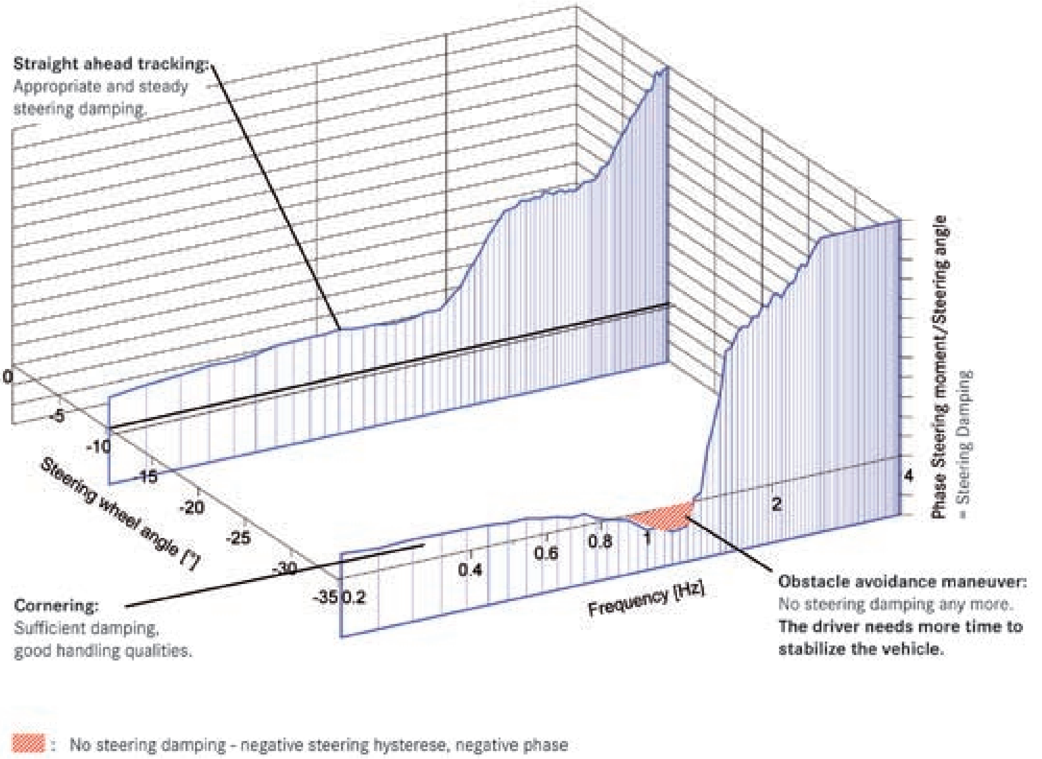
Studying the steering damping as perceived by the driver is comparatively difficult as it consists partially of friction and damping of the steering components. Added to this are the damping of mass via the inertia of the steering wheel, and in particular the considerable influences of the wide range of chassis and superstructure reactions at high speeds.

In conventional procedures, such as, for example the Weave Test [7, 8, 9], evaluations take into account hysteresis loops in which the steering moment overlays the steering angle. However, the various influences of chassis reactions alters the character of the hysteresis loop significantly at different excitation frequencies and steering angles, ❹. In order to obtain a full picture of the steering damping, many hysteresis loops need to be considered at various steering frequencies and steering angles.

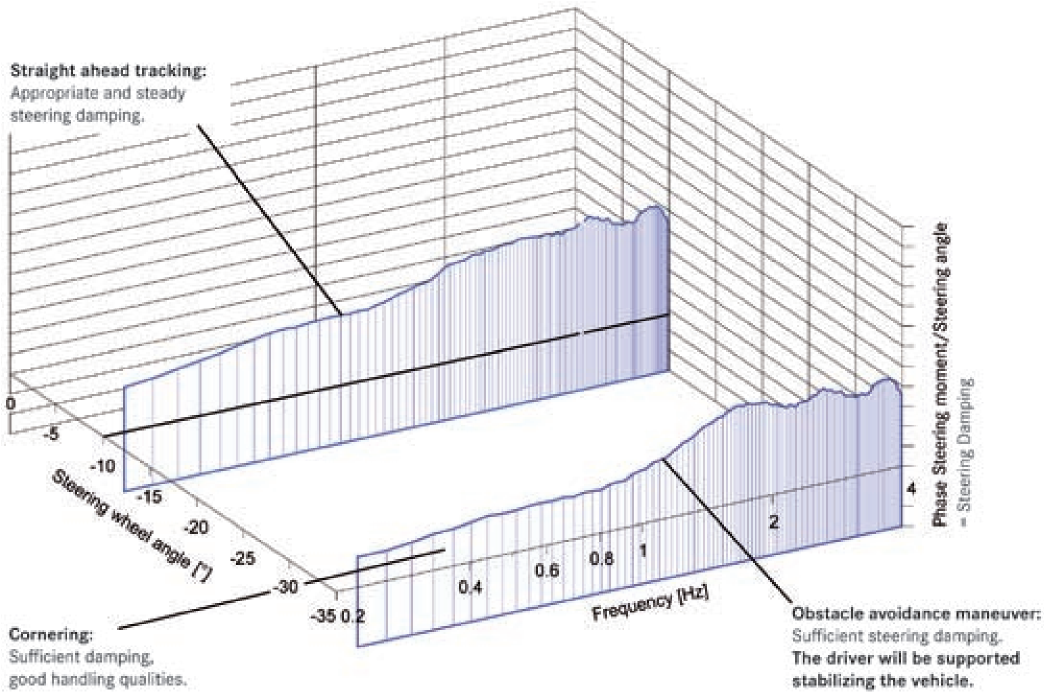
Considering the frequency range, the perceptible steering damping (including frictional influences) dependent on steering frequency and steering angle can be depicted in a single zone, ❺. The steering frequency and the steering angle amplitude is overlaid by the phase delay of the steering moment vis-à-vis the steering wheel angle. It is a measure of the steering damping perceived by the driver. Individual, real-life driving situations can be considered separately in this zone. The areas of low steering wheel amplitude and steering frequencies between 0.2 and approximately 0.8 Hz are ranges in which the driver stabilises the vehicle when driving straight ahead or negotiating generous, gentle corners. The ranges of greater steering wheel angle amplitudes between 0.1 and approximately 0.5 Hz are ranges in which tight and average radiuses are negotiated. The ranges of greater steering wheel angle amplitudes and steering frequencies between 0.5 and 2.5 Hz help the driver to control the vehicle through dynamic driving manoeuvres such as, for example, swerving (0.5-2.5 Hz) and stabilising the vehicle in the process (0.5-1.0 Hz).

In the example of an initial configuration of a vehicle, ❻, the steering damping is located around the steering central position, that is, for instance, for driving straight ahead at an appropriate level which is equally high across the entire frequency range. At higher steering angles, many vehicles display chassis influences in isolated frequency ranges which reduce or neutralise the steering damping. At the same time, the dynamic steering moments fall to virtually zero in this range. As the driver-vehicle control loop prefers the lowest damped frequency in resonance, this comparatively narrow frequency band of, in our case, 0.8-1.0 Hz is essential for the evaluation of the driver in the event of an evasive manoeuvre. Driver feedback on vehicles configured thus are, for example: “Steering centring is too low”, “no clear steering centre position”, “steering feels loose”. Objectively, the drivers require a comparatively long period of time in order to re-stabilise the vehicle after demanding evasive manoeuvres.

An increase in the steering damping in the area of higher steering angle and greater steering frequencies via suitable chassis or steering measures meets the objective set out in the introduction of configuring the reaction of the vehicle in a predictable and understandable fashion. It is also known that drivers in situations of shock tend towards more violent steering reactions as in normal driving situations. In order to reduce a surge of



❻ Steering damping – initial configuration



❼ Steering damping – optimum configuration

the driver-vehicle control loop, a moderate increase of the steering damping in the area of more demanding driving tasks would be sensible.

It has been shown that drivers can stabilise vehicles that display an evenly high damping over the entire driving range more effectively than vehicles with a damping dip within the steering frequency range and higher steering wheel angles. The optimum steering map of the vehicle depicted in ⑥ is shown in ❼.

CONCLUSIONS AND SUMMARY

The reaction of a vehicle to steering movements can be comprehensively described in frequency maps (for example. steering angle, steering frequency). By relating the maps to the driver's seat position, the maps can be used to reliably reflect the actual perceptions of the driver.

Optimisation of the maps, which take into account the limits of human reactions and capabilities, leads to driving and steering behaviour which is better tailored to the driver. The resulting configuration criteria are generally applicable as conventional configuration criteria as they relate less to technical framework conditions of the individual vehicles. It is in principle possible to transfer the configuration targets of vans to heavy commercial vehicles and passenger cars.

Such a tuning of vehicles by means of frequency response maps can effectively help the driver in normal driving situations and support him in his task of controlling the vehicle in more demanding driving situations.

Previous methods of evaluating driving and steering behaviour need to be further developed and perhaps stipulated afresh in consideration of the conflict posed by car-focused driving behaviour and the constraints of van driving dynamics. The reaction of the driver/vehicle pairing in all road situations should be the focus of attention when considering the driving and steering behaviour of vans. Also important to consider are the physical features of a van and the individual habits of the driver as well as the limits of human reaction and capabilities. The method involves recording reactions of the vehicle perceived by the driver using conventional measuring systems and depicting them in frequency response maps. Examined are the essential factors of the steering reaction of the driver and his subjective evaluation of the vehicle reaction. This process makes purely subjectively experienced details of vehicle coordination, the interpretation of which had hitherto only been privy to a small, specialised circle of experts, objectively measurable and thus also makes them available for a comprehensible, structured analysis. It enables greater application of knowledge in the field of control engineering and man-machine systems.

Daimler shows that the frequency response process not only enables the precise configuration of the vehicle characteristics, but can also support the driver in all situations, enabling him to respond in the best possible manner.

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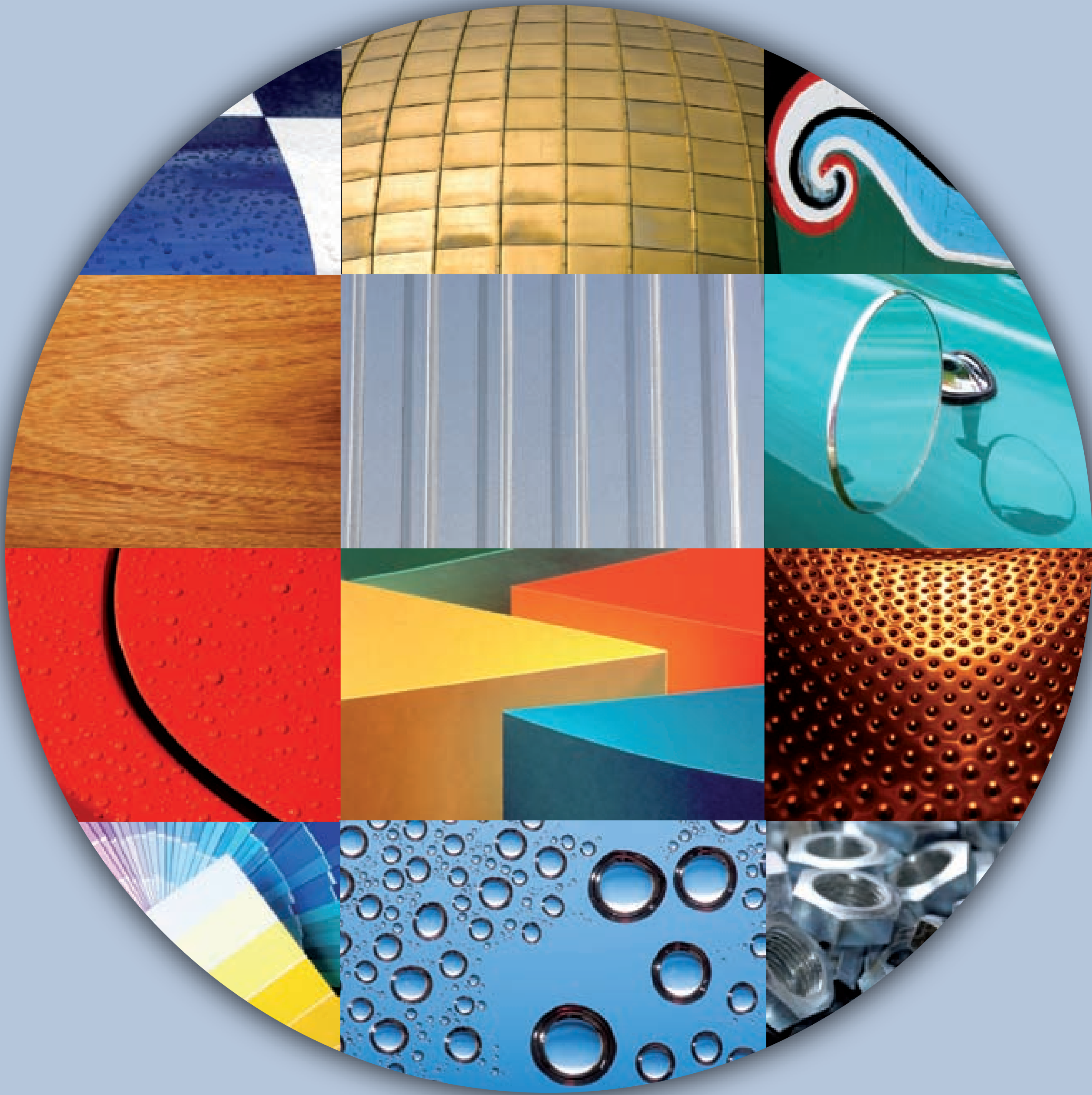

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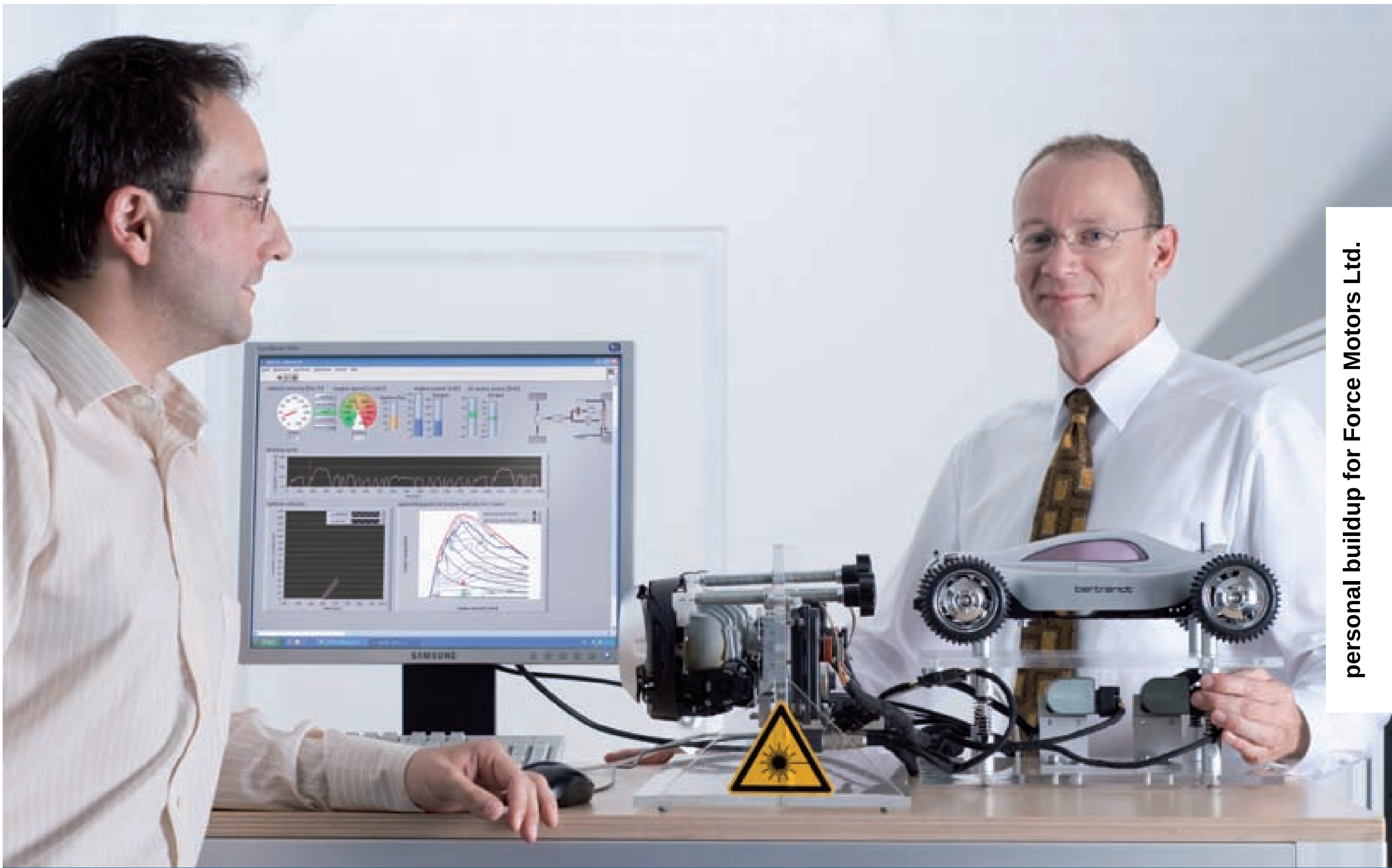

SIMULATION ENVIRONMENT FOR THE ANALYSIS OF DIFFERENT **HYBRID POWERTRAIN CONFIGURATIONS**

A powertrain must fulfil requirements regarding fuel consumption and exhaust emissions and at the same time must be economically viable and meet customer expectations. New concepts with a wide range of topologies are now in the focus of development, thus offering more degrees of freedom but at the same time increasing the time and cost required for analysis. A simulation environment, newly developed by Bertrandt, now makes it possible to analyse different powertrain configurations at an early stage.

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COMBINATION OF MODERN DEVELOPMENT TOOLS

Efficient and effective technologies reduce time, cost and effort during the development process. Only by a combination of modern development tools and technical expertise can the challenging target of lower emissions and lower fuel consumption be achieved while still maintaining an appropriate engine output and offering a high level of driving enjoyment.

Within this area of conflict, the Bertrandt Group has developed a simulation environment that allows different powertrain configurations to be analysed at an early stage in their development. The intention is to evaluate various measures both quantitatively and qualitatively and to include additional information such as navigation data or traffic data. This ability to analyse different powertrain configurations on the basis of fuel consumption, CO_2 and pollutant emissions enables design and series development processes to be supported in a targeted manner.

STRUCTURE OF THE SIMULATION IN THE POWERTRAIN MODEL

The simulation environment "Virtual Powertrain", ❶, is based on Matlab/Simulink. Individual components of the powertrain are represented in sub-models and are linked together via interfaces. The overall model has a modular design, thus ensuring that the sub-models can be modified without influencing other models.

The overall model is designed as a time-discrete backward simulation. This means that the vehicle velocity is determined by the specification of a speed profile such as NEDC, FTP75, US06 or Hyzem. The set velocity of the vehicle is used to calculate the amount of power to be supplied by the powertrain, the fuel consumption, the exhaust emissions and the operating points of the internal combustion engine.

GRAPHICAL USER INTERFACE

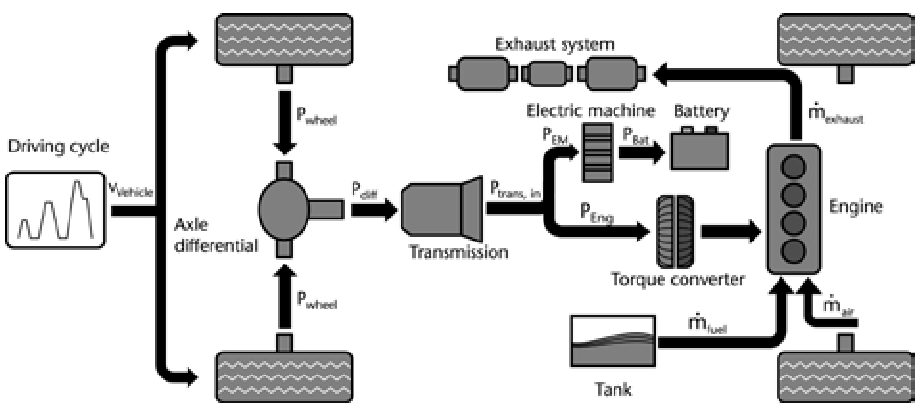
The interface between the user and the simulation environment is a graphical user interface by means of which the powertrain configuration is parameterised (see next section "The Models in Detail") and the simulation is controlled. On completion of the simulation, all output quantities such as the engine power output, the fuel consumption and the brake mean effective pressure are stored as files and graphically visualised.

THE MODELS IN DETAIL

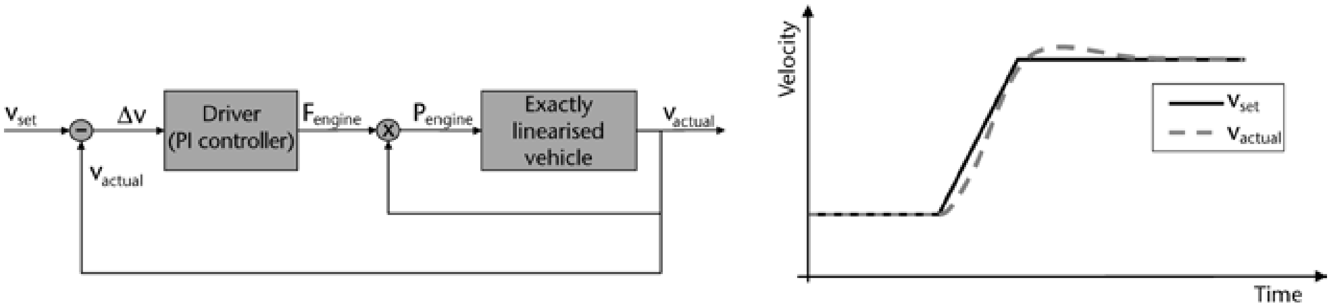
The description of the vehicle environment mainly consists of information such as the road gradient, air humidity, ambient pressure and air temperature. This data is required for calculating the driving resistances.

DRIVING CYCLE

As the first link in the simulation chain, the driving cycle determines the set velocity of the vehicle over time. The specification of a driving profile defined over time forms the basis for the comparison of the fuel consumption and pollutant emissions of different vehicles and powertrain topologies. As vehicle manufacturers are legally bound by the fuel consumption figures determined in predefined driving cycles, these also form the basis for the powertrain simulation. The most important legal driving cycles such as



❶ Structure and components of the simulation environment "Virtual Powertrain"



❷ Controller model "Driver" (left) and resulting qualitative velocity curve during an acceleration phase (right)

NEDC, FTP75, US06 and specific cycles such as Hyzem or Artemis are taken into account as standard in the simulation environment. In addition, one's own individual speed profiles can be integrated.

DRIVER MODEL

The set velocity forms the input quantity of the driver model. By controlling the driving power required by the vehicle, the driver model, which is designed as a PI controller, tries to adapt the actual velocity to the set velocity. In this way, velocities above and below the set velocity are realistically represented, ❷. In order to exclude the influence of the driver on the result of the emissions test or to allow different powertrain configurations to be directly compared with one another, the velocity specification can be exactly represented in the driver model. The weight of the driver is added to the curb weight of the vehicle.

DRIVING RESISTANCES OF THE COMPLETE VEHICLE

Within the simulation environment, all power flows (mechanical, electric) of the complete powertrain are calculated and the individual models of the powertrain are linked with one another sequentially. The starting point of the powertrain model is the determination of the driving power required, which is done by calculating the driving resistances that have to be overcome by the complete vehicle. In order to calculate the total resistance, which is made up of rolling resistance, drag, climbing resistance and inertia, the parameters for the vehicle configuration defined by the user (curb weight including the driver, drag coefficient, frontal surface area, etc.) and the current vehicle velocity and change in velocity are required. The wheel model converts the calculated power requirement into the torque and rotational speed required at the powertrain.

AXLE DRIVE RATIO AND DIFFERENTIAL MODEL

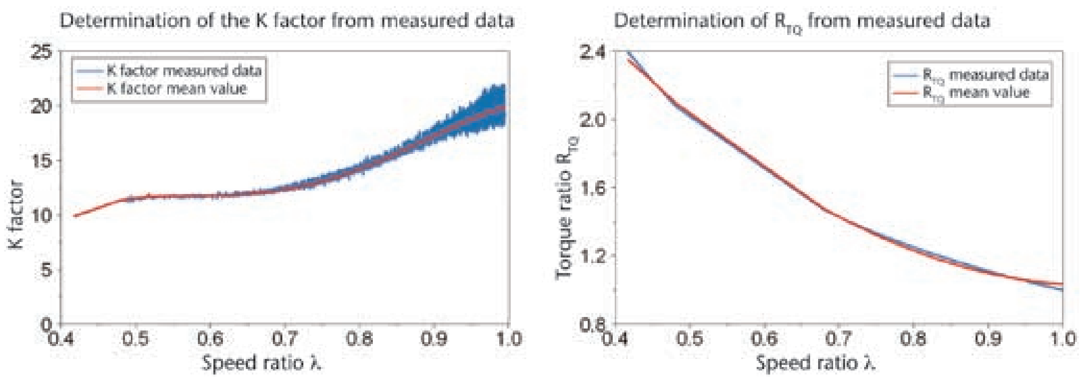
The first conversion of torque and rotational speed in the "Virtual Powertrain" takes place in the differential model, which has been configured with a constant efficiency and an application-specific axle drive ratio. Load-dependent and temperature-related changes in efficiency are considered by characteristic curves and maps.

TRANSMISSION MODEL

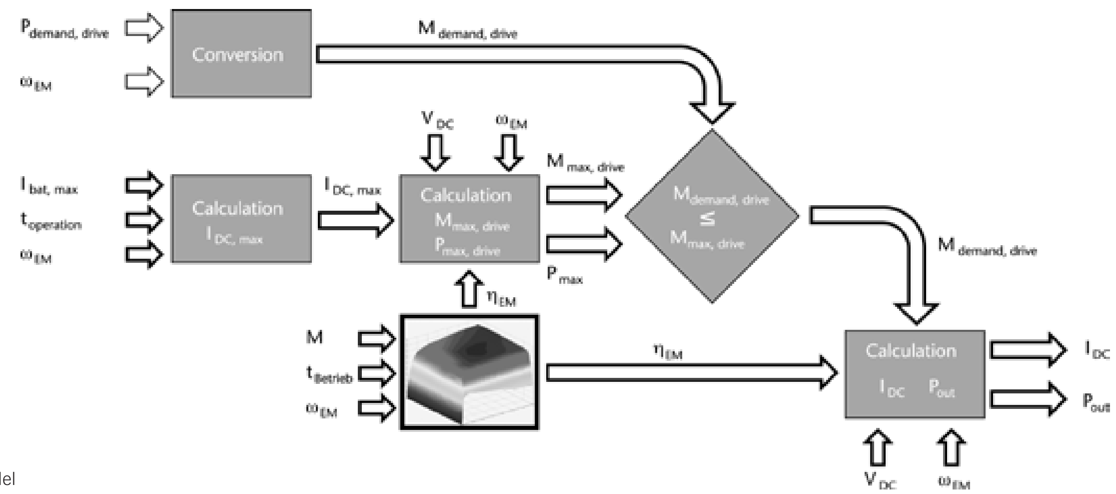
The differential model is directly attached to the transmission model without a drive shaft, which can be configured as an automatic transmission with a torque converter or as a manual transmission with a clutch. In both cases, the number of gears and the corresponding gear ratios have to be defined. Taking into account a power-dependent efficiency, the transmission output speed and torque are converted into a transmission input speed and torque. Temperature-dependent influences are represented by characteristic curves or maps.

TORQUE CONVERTER MODEL

In the simulation of an automatic transmission, a model is additionally activated to calculate the effective engine torque and the engine speed depending on the input and output speed of the torque converter. The characteristic of this torque converter is strongly dependent on the individual



❸ Characteristic curve-dependent modelling of a torque converter in an automatic transmission



❹ Electric machine model during motor operation

vehicle application. Therefore, in the powertrain simulation, each is parameterised by specific characteristic curves or maps. The necessary characteristic quantities can be generated by evaluating transmission or vehicle tests. As an example, ❸ shows the characteristic curves of a torque converter in a series-production vehicle.

INTERNAL COMBUSTION ENGINE MODEL

The engine operating points calculated by the torque converter and transmission models are made available to the internal combustion engine model as input quantities. In this model, the characteristics of the internal combustion engine with regard to its specific fuel consumption, CO_2 emission, etc. are stored in characteristic maps. In addition to the speed-dependent full load curve of the engine torque, the power hyperbolae are defined. For the evaluation of operating strategies to minimise fuel consumption, the speed-dependent power curve for minimum fuel consumption is also calculated.

Due to the change in the engine load and speed of the internal combustion engine over time, both the curves over time and the cumulative values of the fuel consumption and exhaust emissions are approximated. The sub-models of the internal combustion engine include overrun fuel cut-off and idling control, both of which have a significant influence on fuel consumption. Load changes are modelled as an ideal, instantaneous operating point change. Dynamic influences on engine behaviour are not currently represented. Stochastic model approaches for approximating these effects are under development.

EVALUATION AND VISUALISATION

The time curves of relevant output quantities from the individual models recorded during the simulation are submitted to the module "Evaluation and Visualisation". In addition to viewing generally represented results, such as the average fuel consumption or the cumulative pollutant emissions, the user can go on to perform a detailed concept analysis after the simulation. The user interface allows a selective choice of output quantities, which can be evaluated and visualised. Among others, these include the following quantities:

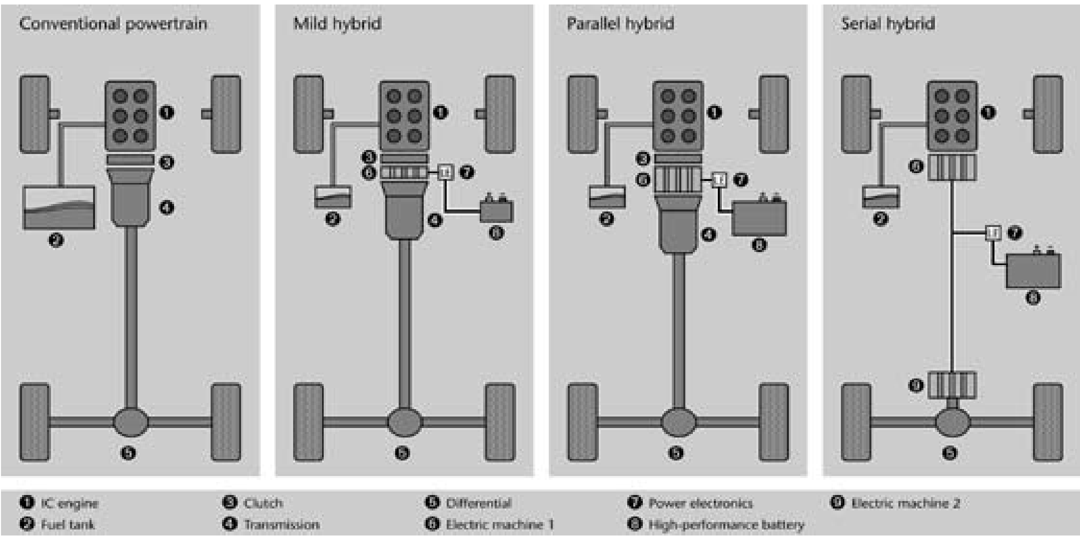
: curves of exhaust emissions and temperatures over time
: curves of power flows, engine speeds and torques over time
: distribution of the load points of the internal combustion engine in the specific fuel consumption map
: frequency distribution of the load points of the internal combustion engine.

Following the specification of the so-called parameter area, the variant analysis option, which can be selected before the powertrain simulation, enables an automatic comparison to be made between different vehicle and powertrain configurations under constant load conditions. The evaluation of the concept is supported by the integrated visualisation option for the simulation results. This option allows, for example, the following variant simulations to be carried out automatically for a fast concept evaluation:

: wheel dimensions
: transmission and axle drive ratios
: engines and application states
: driving cycles
: hybrid concepts and topologies
: operating strategies.

HYBRID POWERTRAINS

The simulation environment "Virtual Powertrain" also offers the possibility to consider electrification functionalities in the powertrain. For this purpose, the basic vehicle is extended by the addition of a battery model and an electric machine model that takes into account the use of the electric machine as a motor and a generator. The power electronics for the conversion, rectification and inversion of the electric voltage is also integrated into the electric machine model. The current maximum torque that the electric machine can provide is calculated in a time-discrete manner during engine operation depending on the rotational speed of the electric machine and the maximum battery current. A comparison between the driving torque required and the available driving torque determines the amount of torque that is to be generated by the electric machine depending on the operating strategy, ❹. The power requirement of electric machine is submitted to the battery model, in which the battery's state of charge is calculated. This model is



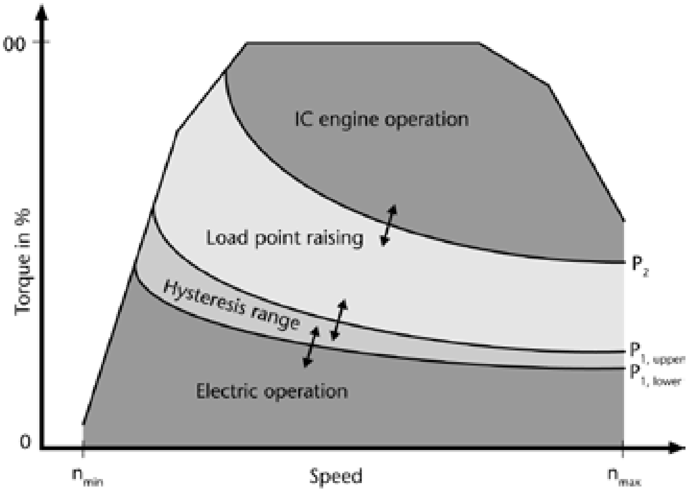
❺ Possible hybridisation functionalities as additions to a conventional powertrain

based on an equivalent capacity and the internal resistance of the battery. The efficiencies of the electric machine when operated as a motor or a generator and the efficiency of the battery are stored in the models as operating temperature-dependent characteristic curves.

The application-specific configuration of the hybridisation in the powertrain is achieved by defining the degree of electrification, the topology and the operating strategy, ❺. Guided by a menu, the user chooses between a micro, mild or full hybrid version, where a micro hybrid corresponds to a start/stop function of the internal combustion engine. For the selection of the mild or full hybrid version, further specifications need to be parameterised. These include, among other things, the maximum power output of the electric motor and the battery capacity.

The operating strategy ensures the intelligent, very complex interaction between the hybrid components. Numerous control approaches are already contained in the simulation environment and new operating strategies can be integrated. By means of powertrain management, the hybrid vehicle can be set up, for example, for maximum range, low pollutant emissions or sportiness. The electrification of the powertrain offers a fuel-saving potential, for example by shifting the load point of the internal combustion engine, ❻, shows a schematic representation of an operating strategy with which the hybrid vehicle is powered purely electrically, electrified with the aim of raising the load point or powered purely by the internal combustion engine. Depending on the battery's state of charge, the power limits P1 and P2 are shifted between the different operating modes. Less fuel-efficient operating points of the internal combustion engine are therefore avoided and the additional electric drive system is effectively employed.

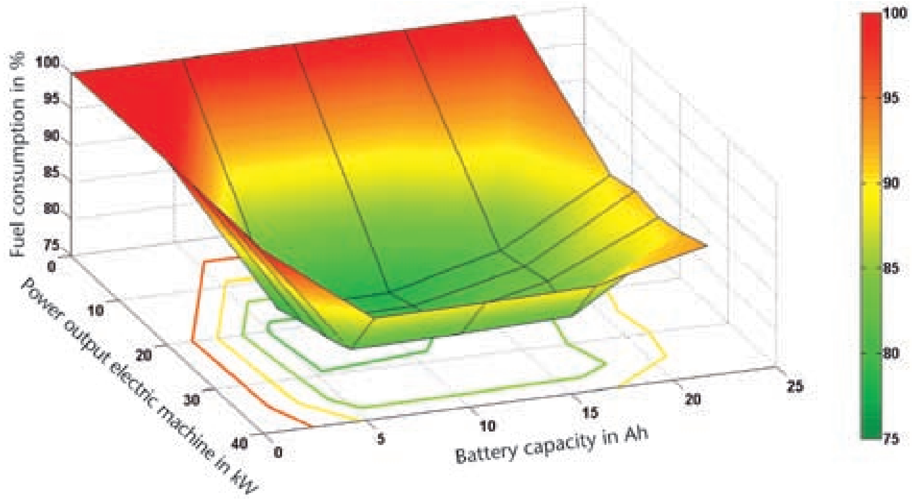
The modular design of the simulation environment simulation allows the operating strategy to be extended, for example to consider traffic flow or topographical data. This predictive powertrain control provides a predictive calculation of the most efficient means of propulsion. Initial approaches already consider topological data in the powertrain control system. For this purpose, a separate cycle with route and height information was developed.



❻ Example of an operating strategy for a hybrid vehicle application

ELECTRIC VEHICLES

In current powertrain development, hybrid technology represents the transition from internal combustion engine drive systems to electric drive systems. As a key technology in this development, the energy density of the batteries is the bottleneck in the complete electrification of the powertrain, as fully electric drive systems still have a limited driving range and their weight is too high. However, due to the high overall efficiency of the drive system and in view of the increasing depletion of fossil fuels, electric drive systems are a promising alternative to conventional drive systems in urban applications.



❼ Determination of the electrification combination of an electric machine and a battery to achieve optimum fuel consumption

Therefore, the powertrain model also allows the simulation of electric powertrains. The driving power required in a driving cycle is generated by one or more electric machines supplied with energy from one or more batteries. In the sample application of an electric vehicle with wheel hub motors, the power required is distributed over two or four electric machines in the powertrain simulation. Wheel hub drive systems do not require a drivetrain and drive shafts and the mechanical power flow is reduced to the electric machines and the wheels.

ELECTRIFICATION OF AN SUV

In an application example, an SUV (equivalent mass bigger than 2000 kg) with a diesel engine and a conventional drivetrain is extended by the addition of a parallel hybrid configuration. In order to analyse the optimum combination of an electric machine and a battery, a simulation is carried out on the basis of a pre-defined operating strategy with the primary aim of achieving minimum fuel consumption without at the same time evaluating the driving dynamics, ❼. For this application example and the operating strategy on which it is based, a reduction in fuel consumption by 19 % compared to a conventional powertrain (corresponding to 100 %) is possible. This requires an electric driving power of 20 kW and a capacity of the lithium-ion battery in the range of 8 Ah. The simulation takes approximately four hours.

A comparison of the load points of the diesel engine in the engine map shows the optimisation potential of the parallel hybrid configuration.

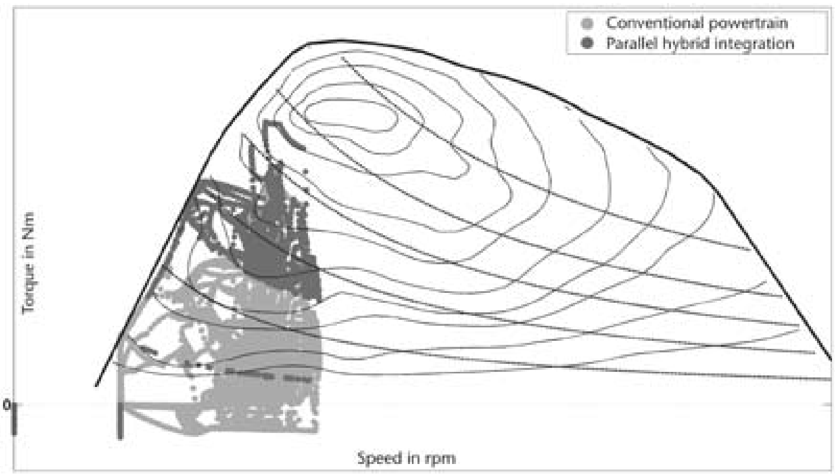
Apart from purely electric operation, the internal combustion engine is operated in load ranges with a lower specific fuel consumption (load point raising), ❽.

SUMMARY

Development service provider Bertrandt has developed its own modular and flexible software environment called “Virtual Powertrain”, which provides extensive possibilities for analysis and configuration in powertrain development. Initial validations with comparative data from real vehicles confirm the qualitative and quantitative information value of the simulation with appropriate computing power.

In addition to future challenges in conventional powertrain technology, the degree of complexity in powertrain development will rise significantly due to the increase in the numbers of hybrid and electric vehicles.

Powerful and efficient development tools will support these development activities to a greater extent than before. With this new “Virtual Powertrain” software environment, Bertrandt is going a step further as a developer in helping to design tomorrow's mobility.



❽ Comparison of the operating points of the internal combustion engine in the engine map for different powertrain configurations in the NEDC

MODULAR AIR QUALITY SYSTEM FOR INTERIOR COMFORT

As traffic volumes and the associated burdens on drivers increase, the need for a relaxing, rejuvenating atmosphere in the vehicle becomes ever greater. The five-stage air quality system from Behr can meet these requirements in full, right down to generating a truly experienceable atmosphere of wellbeing in the vehicle interior of passenger car and truck, with ionized air and a selection of fragrances for every preference.

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INTRODUCTION

High air quality in the vehicle is possible only through the implementation and interplay of the technologies detailed in the “air quality steps” in ❶, which adopts and promotes a bottom-up approach to meet the requirements. For instance, ionization and fragrancing only make sense if the air has already been largely purged of particles, harmful gases, odors, and to some extent microorganisms, that means through filtration, sensor technology, the evaporator surface materials BehrOxal or BehrOxal nano.

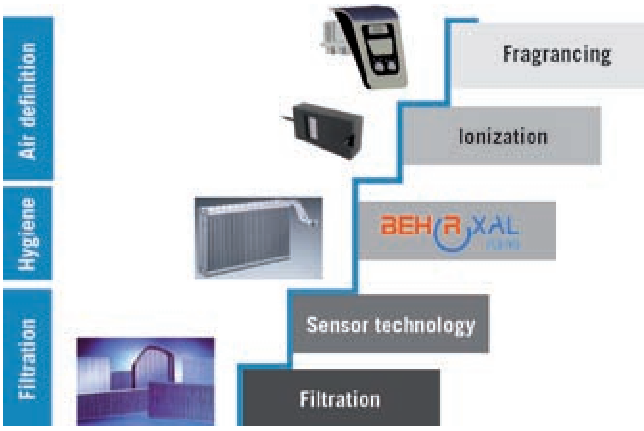
The primary focus of this article is on air ionization and fragrancing. The preceding stages of the air quality steps are only briefly addressed.

FILTRATION, SENSOR TECHNOLOGY, BUT ALSO BEHROXAL AND BEHROXAL NANO

Modern vehicle cabin filters, pure particle filters, and hybrid filters (particle filters with active carbon layer), are designed in accordance with DIN 71460. The particle collecting properties of particle filters are analyzed using test dusts, while the absorptive capacity of hybrid filters is examined using test gases.

Harmful gases (CO, NO_x) as found in diesel and gasoline exhaust gases, for instance, are detected using air quality sensors. If the concentrations of these gases increase substantially, the signal transmitted by the sensor prompts the climate control system to switch automatically to recirculated air mode to avoid overloading the filter and to prevent a filter breakthrough with harmful gases entering the vehicle cabin.

The BehrOxal surface treatment modifies the aluminum surface of evaporator, giving it hydrophilic properties. Any accumulating condensation forms a thin, rapidly draining film of water purging impurities together with microorganisms and their nutrients from the surface instead of producing large slowly drying water droplets. The optimized evaporator design promotes the draining of water, enabling the surface to dry more quickly after operation. This, in turn, reduces the prevalence of microbial growth, for which water and nutrients are required. It is also possible to apply biocide coatings to the evaporator surface.



❶ Steps to high air quality

However, field studies have revealed that the application of conventional biocide coatings is largely ineffective against odor-producing germs. Furthermore, these products are quickly washed out by condensate, meaning that any effect they have is short-lived at best. The new BehrOxal nano coating has been developed for this reason. This consists of a polyurethane lacquer coating containing an abrasion-proof and leach-resistant nanoscale biocide that destroys odor-producing germs.

IONIZATION

The ionization of air, a widespread practice in households in Asia, is becoming more widely used in motor vehicles as well. However, simple modification of domestic ionization equipment is not sufficient to meet the stringent requirements of the automotive industry in terms of reliability, operational safety, and the minimization of ozone production. For this reason, Behr, together with the world-renowned electronics supplier Samsung, as development partner, has developed an ionization module specially for automotive applications. This module, ❷, generates only a negligible amount of ozone. The device enables the vehicle occupants to choose between two operating modes (Clean and Relax) that produce different ion types in high concentration.

CLEAN OPERATING MODE

The functional principle of the "Clean" mode involves the generation and application of high voltages to a ceramic cathode and needle-shaped anode using a special electronic system. First, hydrogen cations are generated on the cathode from water vapor in the air, and these are subsequently reduced to atomic hydrogen at the anode. Superoxide ions (O2-) are also formed at the anode and react with the hydrogen atoms to produce hydrogen peroxide ions (HO2-). These anions settle on microorganisms present in the air (bacteria, viruses, fungi, or spores) and deprotonate their protein envelope, to a certain degree impairing their biological function. The microorganisms, although still present in the air, lose their pathogenic effect. They are "deactivated" as it were, since the damage to their protein coating

❷ S-Plasma ionizer module from Behr/Samsung for vehicle interiors

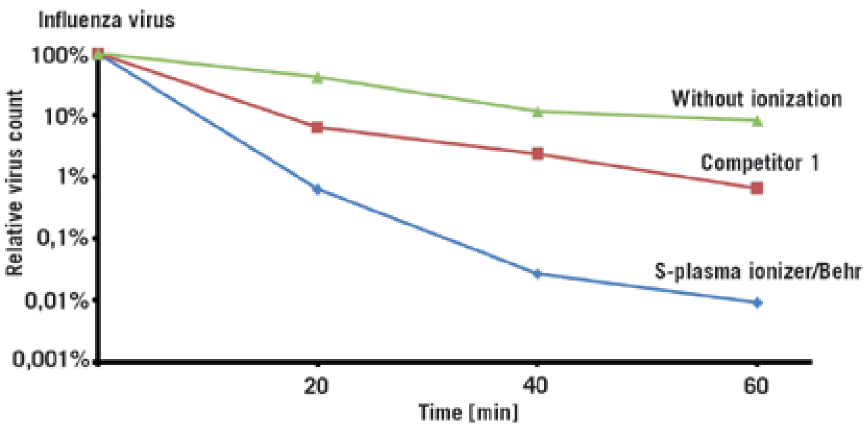
removes their capacity to invade human cells.

A renowned biotechnical institute demonstrated this practice of "deactivation" using the bacterium Staphylococcus aureus as an example. In the laboratory experiment, the count of this bacterium was reduced by 97 % within 20 min using ionization and the product described above. The number of influenza viruses dropped to one thousandth (0.01 %) compared with the virus count without ionization (10 %) in just one hour in the test chamber. A competitor product tested in the same way achieved a reduction only to approx. one tenth (1 %), ❸.

❹ summarizes the efficacy for various pathogens and allergenic agents examined by external institutes.

However, determining efficiency under laboratory conditions is not enough in itself. For instance, tests conducted on a used vehicle (two air samples taken daily, over six consecutive days) revealed a reduction in molds by up to 90 % (average 65 %) and in bacteria by up to 60 % (average 35 %), ❺. This shows that the technology is also effective in a real vehicle environment, even though test conditions in this respect are much more difficult and less well-defined than in a laboratory.

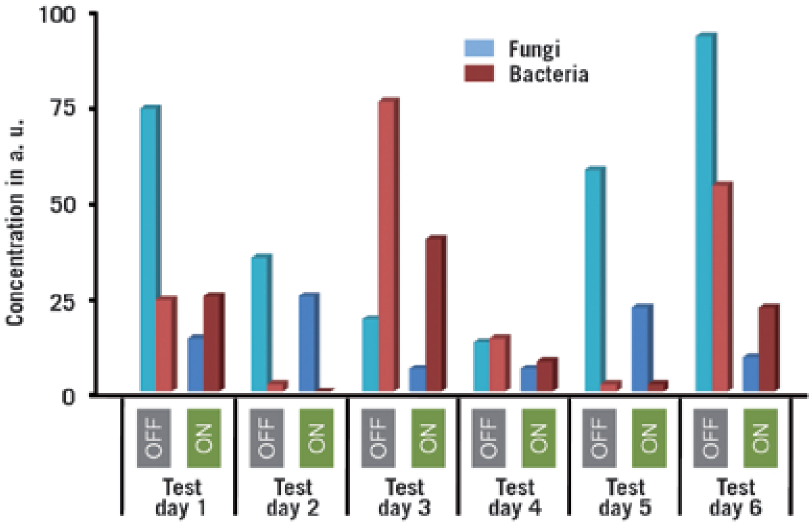
Additionally, the intensity of odors produced within the vehicle, such as from



❸ Concentration of airborne influenca virus A (H3N2) with the operation of the S-Plasma ioniser, competitor product, and without ionization – the efficacy test was conducted in a 1 m³ chamber in the department of biotechnology of Yeonsei University, South Korea

MICROORGANISMS / GERMS	EFFECTIVENESS	CONFIRMED BY
Influenza virus type A (flu)	99,7 % / 99,6 %	Kitasato Environmental Science Center / Yonsei University
Corona virus (Sars virus)	> 99 %	Kitasato Medical Center
Yellow micrococcus, coliform black mold, green mold	99,9 %	Korea Consuming Science Research Center
Allergenic agents from house dust mites / cats / dogs	Effectiveness verified	British Allergy Foundation (BAF)
Methicillin-resistant Staphylococcus aureus (MRSA)	99,9 %	Kitasato Environmental Science Center

❹ Efficiency of S-Plasma Ionizer on various bio-contaminants as confirmed by collaborative research center



❺ Average concentration distribution of airborne bacteria and fungi, measured in the automobile; the operation time of S-Plasma ionizer was 30 min, tested in the R&D center of Samsung Electronics

IONIZATION SYSTEM	BEHR AUTOMOTIVE	COMPETITOR AUTOMOTIVE	NON-AUTOMOTIVE	LOW-COST AUTOMOTIVE
OZONE CONCENTRATION [ppm]	1.2	2	3	4
NEGATIVE ION CONCENTRATION [1/cm³]	5×10^6	3×10^6	4×10^6	2×10^6

❻ Comparison of various ionizers

cigarette smoke, is reduced. And in trials, involving two groups of participants, each with 50 persons, fewer cases of eye distress were recorded in the group with ionization as compared to the control group without ionization. These trials were conducted in an urban traffic environment as early as the 1990s. Neither of the two groups knew about the ionization. The evaluation also recorded a significant increase in driver vigilance in test drives featuring ionization. The report concerning these trials concluded with the following sentence: "Thus existing thermal and air quality conditions in vehicles have measurable effects on the productivity of drivers" [1].

RELAX OPERATING MODE

The "wellness effect" produced by the air revitalization of the "Relax" mode is based on an increased concentration of negatively charged ions (oxygen anions). Their positive effects on vehicle occupants have been demonstrated through observation and surveying of participants in test drives, and through psychological and medical testing. The results indicate a statistically significant improvement in reaction time tests. The participants reported feeling more capable, and found it easier to concentrate than in similar controlled situations without oxygen anions [2]. Medical examinations also showed reduced levels of the stress hormones cortisol, serotonin, adrenalin, and chromogranin A in the blood, urine, and saliva [3, 4].

The improvements are dependent on the individual sensitivity of the probands and the concentration of negatively charged ions achieved through ionization. ❻ provides a comparative summary of data produced for various ionization units under standardized conditions on an in-house test bench. The data clearly indicate that, at 5×10^6 ions per cm³, Behr's own unit achieves the highest concentration of ions with a minor amount of ozone. For comparison, the following values represent naturally occurring concentrations of negatively charged ions (1 per cm³):

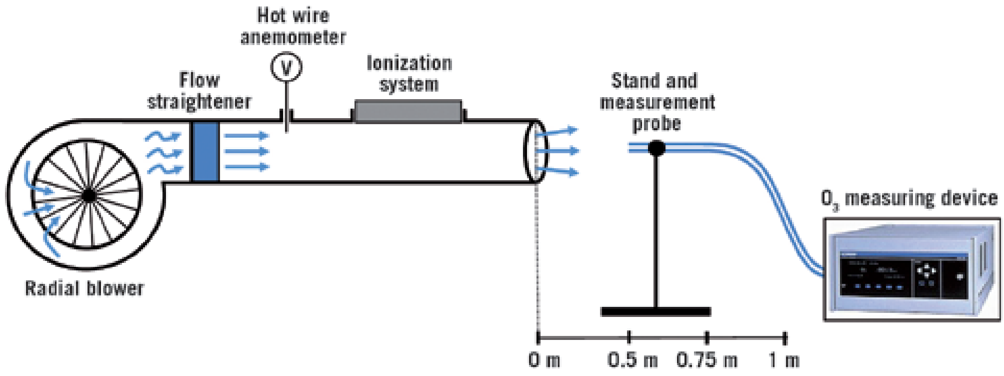
- waterfall, forest: 50,000
- mountains, coast: 5000
- rural areas: 700 – 1500
- inner-city parks: 400 – 600
- urban sidewalks: 100 – 200
- enclosed spaces: 0 – 200.

The ion concentrations recorded in the vehicle (driver/passenger head area) with ionization correspond to those of mountain and coastal regions. Compared with measurements of ion concentrations taken in otherwise non-ionized vehicle cabins, this represents an enhancement by a factor of 100, meaning a significant increase in the concentration of negatively charged ions is actually possible in vehicles.

SAFEGUARDS AGAINST OZONE FORMATION

⑥ also shows that the Behr module exhibits the lowest ozone level. This is achieved using a specific combination of electric high voltage, a tailored electrode geometry, electronic control, and coordinated, aerodynamic integration. The result is a maximized ion count with a minimum of ozone. ❼ illustrates the application oriented test assembly used to produce the laboratory measurements. Standard distances between the outlet of ionized air at the vents and the heads of the front-seat passengers are deemed to be in the 50 to 80-cm range. ❽ shows that, at a distance of 50 cm between air outlet and measurement probe, the increase in ozone concentration due to the process of ionization (measured in "Clean" mode) is extremely low, and is far below the odor threshold (40 µg/m³). Values at distances of 75 and 100 cm are close to the limit of detectability, and exhibit no difference between activated/deactivated ionizer.

In-vehicle measurements have verified these laboratory results, recording an entirely harmless ozone concentration in the vehicle interior (likewise measured in "Clean" mode). This value, produced with doors closed and ionization system activated, is close to the detection limit of the measurement device (0.5 ppb), and is therefore significantly below values for external air, as illustrated in ❾. Figures on the day were more than a factor of 10 below the public ozone warning level (240 µg/m³). In results of a 24-hours measurement conducted in recirculated air mode, the ozone level also remained at the limit of detectability. These findings verify the absence of ozone enrichment in



❼ Laboratory measurement setup

the vehicle cabin, even with extended duration of ionization. Use of the ionization module therefore has no relevant impact on in-cabin ozone concentration.

APPLICATION OF THE IONIZATION MODULE

It is vital for the effectiveness of ionization that airflows are considered when integrating the ionization module into the HVAC system. Analyses with computational fluid dynamics (CFD) are used to ensure that the incidence of transverse flows and backflows between electrodes are kept to an absolute minimum. This is the only way to safeguard the formation and delivery of HO_{2^-} ions in quantities sufficient to effectively “deactivate” microorganisms in the air.

FRAGRANCING

A study conducted by the German market research institute Gesellschaft für Konsumforschung (GfK), probing the market potential for integrated fragrancing systems for passenger car and truck interiors, concluded the following.

In the subcompact and compact segment, 75 % of the participating passenger car drivers/owners surveyed considered a fragrancing system (a Behr prototype was presented) to be an interesting or highly interesting product. The total surveyed (Germany) was n = 400. In the mid-range segment, this rose to 83 %, rising again in the premium segment even as much as 87 %. Approximately 60 % of those surveyed were reported to be already using one of the many popular scented air freshening attachments/air freshener trees. For the truck drivers the total surveyed (Europe) was n = 300. More than 80 % of truck drivers questioned similarly indicated an interest in fragrancing systems.

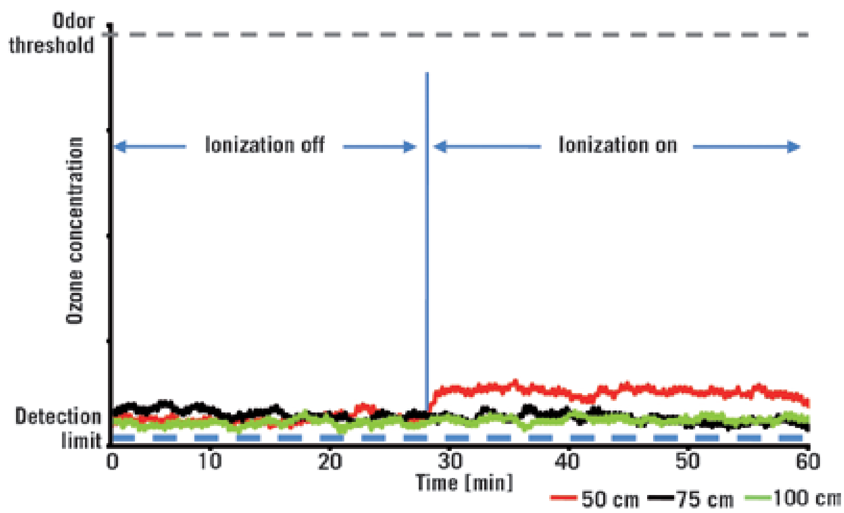
FRAGRANCING SYSTEM CRITERIA

In recognition of this high interest, as well as in attempts to further customize and enhance passenger comfort, the first automotive manufacturers have begun to offer fragrancing systems in their vehicles as standard and optional equipment. An integrated fragrancing system must adhere to the following principles, also supported by market research:

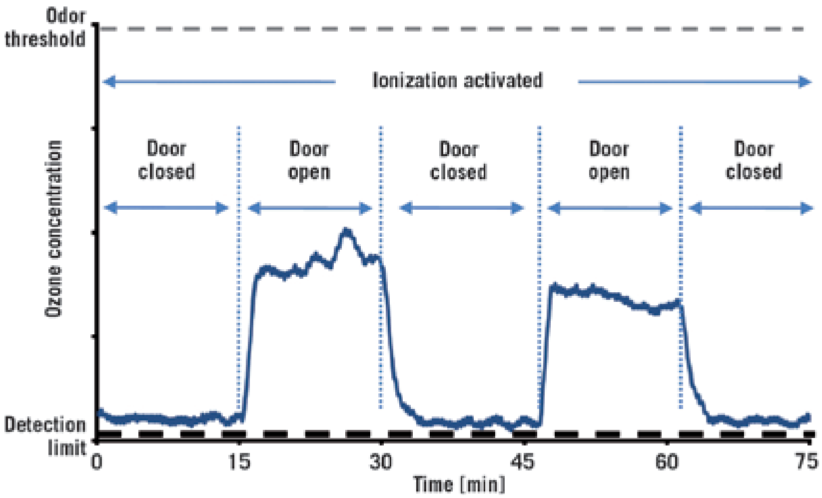
- A sufficient number of fragrances to suite individual preferences must be available; occupants must have the option to choose between them.
- The system must not pose any health hazard, and must be suitable for people with allergies.
- The fragrance intensity must be adjustable.
- Fragrance diffusion must be regulated to ensure occupants do not become accustomed to the fragrance and then no longer notice it.
- Operation must be user-friendly,and cartridges must be quick and easy to replace.
- Fragrancing system and operation must be styled to reflect the high quality of the vehicle.

Behr’s own fragrance diffuser, the first fully integrated system with an autonomously operating air supply, ❿, meets all of the above requirements.

Preferences and compatibility: There are currently various fragrances and fragrance intensities available to customize the scenting experience. New compositions are also being developed as automotive manufacturers and perfume manufacturers join forces. All substances used in fragrances comply with relevant health



❽ Comparison of ozone concentrations – at various distances from the air outlet vent (laboratory conditions)



❾ Comparison of ozone concentrations – with open and closed doors

and environmental protection regulations. If required by the customer, allergenic substances can be excluded or explicitly identified on the packaging. Only harmless substances that have been appropriately tested and approved for commercial sale are used.

Fragrance intensity and accustomization: Everyone experiences scents differently, according to personal emotive associations and memories. The ability to manually adjust the fragrance intensity in three stages caters to these personal differences. Interval-based operation prevents accustomizing to the scent. The consequence of accustomization is a decreased and later entirely diminished awareness of the fragrance. The olfactory mucous membranes in the nose become fatigued with the prolonged onset of the fragrance.

If an individual remains in a scented environment for extended periods, the degree of awareness of the particular fragrance is lost. The ability to detect other odors remains, however. The Behr module therefore provides a controllable pulse mode with intervals, to prevent diminished awareness and prolong the service life of the fragrance cartridges. The system is not intended for obligatory scenting; the user is always free to select the fragrance and its intensity, when it should be switched on, and its duration. Furthermore, the user can pause or entirely deactivate the fragrancing system at any time.

Operation and cartridge replacement: The scenting module includes fixtures for two mutually connectable fragrance cartridges, each of which can be replaced quickly and easily by way of a "push-push" action, just like a cup holder. There is no risk to the user of contamination from the use of these cartridges. This method of operation ensures intuitive and error-free handling.

Styling: An adaption to the high quality dashboard design is possible with a customer specific front panel of the fragrancing system. Therefore, the fragrancing system's basic module has not to be modified.

FUNCTIONING OF THE FRAGRANCE DIFFUSER MODULE

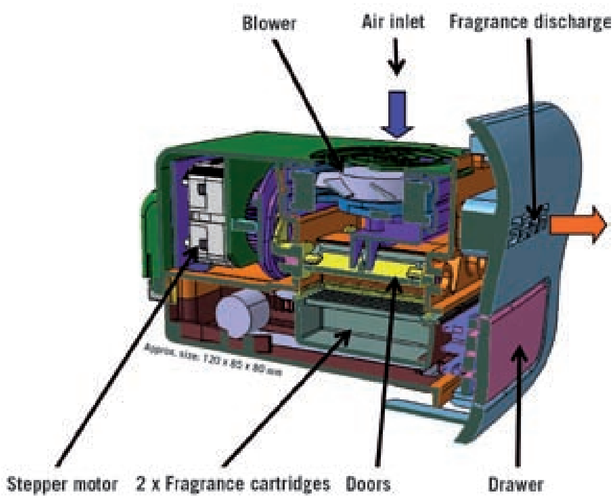
The air to be scented is drawn from the conditioned vehicle cabin using an integrated blower in the fragrance diffuser module, ⑩. Air ventilation doors direct the flow of air across the fragrance cartridges and back into the vehicle interior. The fragrance intensity corresponds to the degree at which the air is enriched with fragrance molecules. The fragrance diffuser is controlled by an independent control head that receives all necessary parameters regarding climate control settings from the air conditioning control unit.

These parameters include air temperature and air distribution in the cabin (ventilation level, footwell), operating mode (external or recirculated air mode), and airflow through the cabin. The following can also be considered: the duration of use of the fragrance diffuser (high fragrance intensity for short journeys, low intensity for longer journeys) and the intensity of the fragrance used. Based on this information, a preset algorithm is used to calculate the pulse amplitude, pulse duration, and interval time. It is therefore possible to regulate the airflow across the fragrance cartridges in the module to ensure the preselected intensity remains the same, even though vehicle and cabin parameters are changed.

FRAGRANCING VALIDATION

The selection and validation of the fragrance precede the validation of the module. If the fragrance's acceptance, stability, and lifespan are verified, the module is validated. Two factors are particularly important in this regard:

: a consistent fragrance intensity, despite modification of air conditioning system settings



❿ Layout of fragrance diffuser module from Behr

: material compatibility of fragrance substances and module materials, and, if necessary, peripheral module equipment such as air ducts and vents in the dashboard.

Olfactometric analyses have proven inadequate with regard to evaluating the consistency of fragrance intensity. Behr specialists have therefore developed a procedure to facilitate precise quantification of scent molecules in the vehicle interior using a gas chromatograph and/or flame ionization detector (FID). Since perfumes are highly complex mixtures, sometimes comprising over 100 separate components, the process uses test fragrances as model mixtures. These consist of non-volatile, medium and high boiling components. Their "fingerprints" are recorded in the analysis. These model mixtures enable the derived quantities to be tracked. The degree of airflow scenting lies in the lower ppm range, depending on the door opening angle and duration. The objective is to keep the degree of fragrancing within the detection threshold of the vehicle occupants at all times, since only in this way can a pleasant scent experience be maintained over extended periods.

Behr also has expertise with respect to the determination of material compatibility. This concerns the selection of materials and components for the fragrance diffuser module, air ducts, and outlets.

Necessary materials testing of surrounding components is highly coordinated, involving automotive manufacturers, fragrance manufacturers, and manufacturers of components that come into contact with fragrance substances (including essential oils and solvents), both inside and outside the fragrance diffuser. This means that the vehicle in which the diffuser is installed must also be included in the validation process.

SUMMARY

The holistic approach by Behr is essential for the generation of high customer benefit. Therefore, the supplier uses a bottom-up approach with so-called air quality steps. To do so requires each step of the air quality – filtration, sensor technology, surface treatment as BehrOxal and biocide coating as BehrOxal nano, ionization, and fragrancing – to be customized to the vehicle. With this air quality system, future customer requirements can be better addressed and greater comfort brought to the vehicle interior as an environment and workplace.

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Therefore, since the No. 4 issues of 2008, ATZ and MTZ have the status of refereed publications. The German association "WKM Wissenschaftliche Gesellschaft für Kraftfahrzeug- und Motorentechnik" supports the editors in the introduction and execution of the peer review process. The WKM has also applied to the German Research Foundation (DFG) for the magazines to be included in the "Impact Factor" (IF) list.

In the ATZ|MTZ Peer Review Process, once the editors has received an article, it is reviewed by two experts from the Advisory Board. If these experts do not reach a unanimous agreement, a member of the Steering Committee acts as an arbitrator. Following the experts' recommended corrections and subsequent editing by the author, the article is accepted.

THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN ATZ.
ATZ PEER REVIEW
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

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HYBRID STEERING FOR REDUCING FUEL CONSUMPTION OF COMMERCIAL VEHICLES

During the course of a development project undertaken by the Truck Product Engineering department at Daimler AG in close collaboration with Karlsruhe Institute of Technology (KIT), a methodology was developed for modelling and validating a reduced fuel consumption hybrid steering system for trucks. To establish an optimum system design and to evaluate a variety of technologies for the hybrid steering system, a development methodology approach was put forward. This will allow the developer to make a detailed analysis of the complexity of this mechatronic system, to quantify the optimisation potential and to draw up a requirement profile for the simulation and experiment-based development environment, taking particular account of modelling quality and development efficiency.

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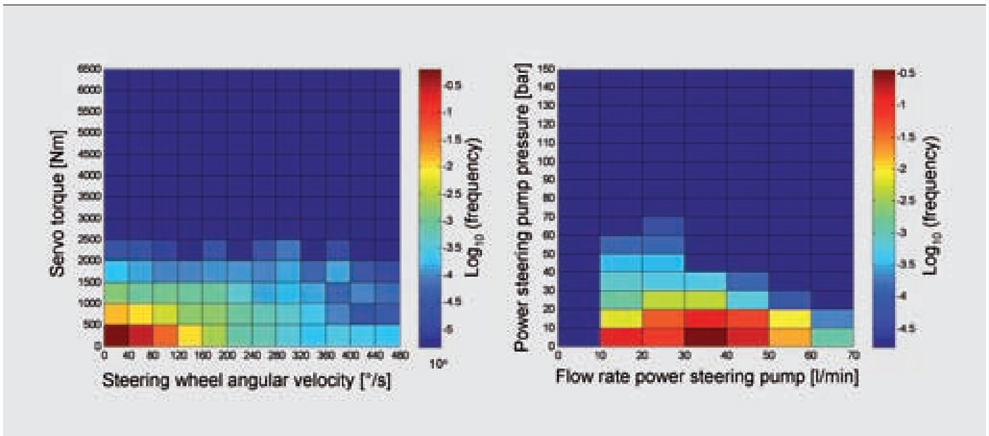
1 INTRODUCTION

According to the BGL (Federal German Association for Road Haulage, Logistics and Disposal), fuel costs, at 26 % of total costs, represent the greatest cost factor after labour costs [1]. This means that optimizing the energy-efficiency aspects of auxiliary assemblies and sub-systems is acquiring ever increasing importance in the commercial vehicle sector. There is a great potential to increase energy efficiency in the hydraulic open centre steering system (HPS-OC) used today in commercial vehicles.

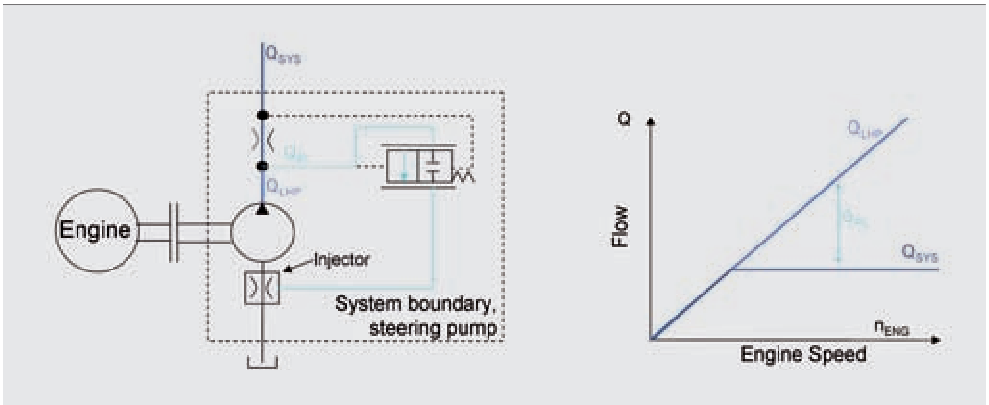
In order to evaluate the energy flows in the hydraulic open centre steering system in current use, the degree of mechanical power assistance was measured on test runs by a 40 t semitrailer truck equipped with pressure sensors. The power steering pump's mechanical input, which, together with the power steering assistance, is strongly influenced by the individual flow losses in the steering system, was determined by precise measurement of the individual steering components on a purpose-built hardware in the loop (HIL) test stand in the research department at Daimler AG.

Using the recorded test run data, ❶ illustrates the distribution of the parameters for servo torque and steering wheel velocity, parameters, which are decisive factors for the hydraulic servo power. The distribution of the parameters for power steering pump pressure and delivered volume, parameters that characterise the power steering energy input, are also shown. A theoretically-calculated comparison shows that the degree of mechanical power assistance used represents just approximately 1 % of the power input.

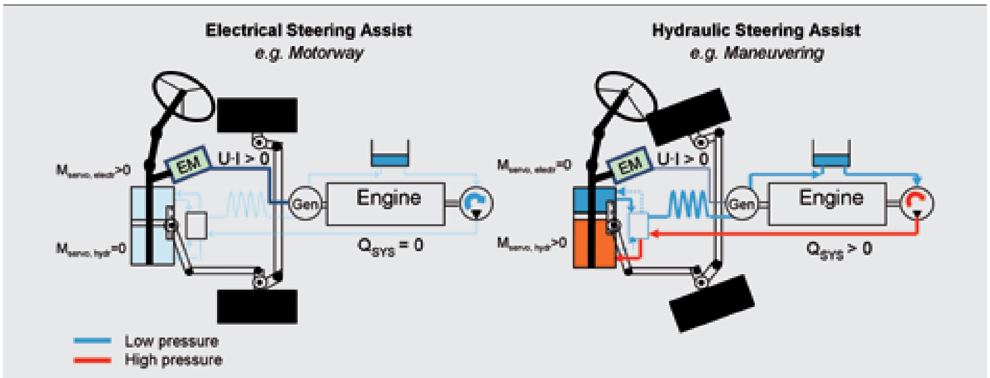
The high losses are attributable to the fact that the traditional power steering pump generates a delivered volume dependent on the engine speed and feeds it against the system pressure, irrespective of the actual steering requirement, ❷. Since the steering system is designed to meet the criterion for maximum steering speed when turning the steering wheel while the vehicle is stationary, this results – even at engine idling speeds – in the requirement for a relatively high pump displacement volume [2]. At higher engine speeds, the system volume flow is limited, when the residual delivered volume being ducted into a return run inside the pump [3]. The effect of this, particularly for long-haul commercial vehicles, is a constant hydraulic power loss due to the long distance element in their overall mileage.



❶ Collective load for a servo steering pump of a 40-t semitrailer tractor



❷ Principle of the servo steering pump



❸ Mode of action of the consumption-reduced hybrid steering system: pure electric steering for example on a motorway trip (left), hybrid steering for example during manoeuvring (right)

2 HYBRID STEERING SYSTEM

Reducing the power input in passenger cars has been achieved through the electrification of the steering system, either by partially electrifying the hydraulic steering system, using an electric motor to drive the power steering pump, or by completely replacing the hydraulic power steering with an electromechanical adjusting motor [4]. Because the high steering forces and power inputs of up to 6 kW in commercial vehicles do not make electrification an easy with the current design of on-vehicle power supplies, the innovation of a hybrid steering system represents an attractive solution. Compared with the conventional steering system, the hybrid steering system is complemented by an electrical steering adjusting mechanism and a positive electrically controllable volume flow adjustment mechanism. It offers, on the one hand, the potential for a major reduction in power and, on the other, the possibility of realising steering assistance functions.

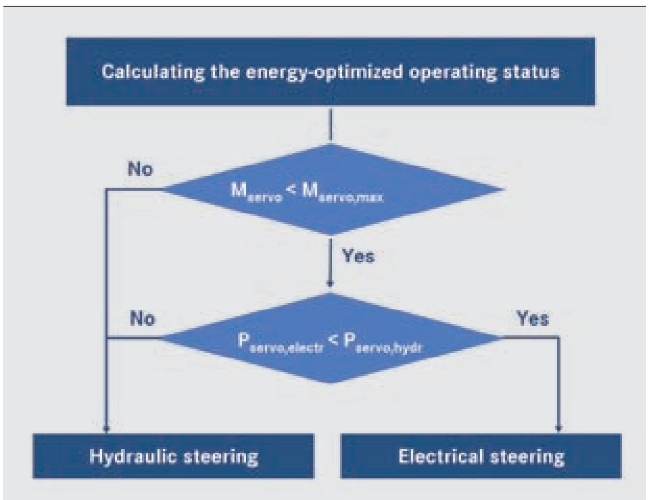
❸ illustrates the principle underlying reduced consumption steering. The basic idea is that in driving situations where only very little or no use is made of the steering wheel, steering assistance can be provided by the electric adjusting motor and, at the same time, the hydraulic losses can be reduced by requirement-led restricting of the hydraulic system volume flow. Only in driving situations, which require a great degree of power assistance would power steering be provided by assistance from the hydraulic sub-system.

With the objective of achieving optimum performance for the mechatronic system as a whole, a power-led operation strategy was developed within the framework of the development project. Such an operation strategy makes it possible for power steering to be provided with minimum energy use always being the most important consideration, ❹.

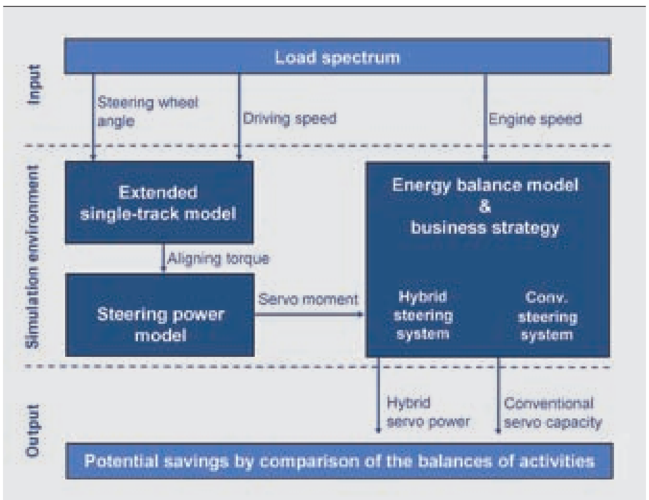
At a first decision stage, by comparing the demand for servo torque detected by the torque sensor with the maximum possible electrical servo torque, a decision is taken as to whether power assistance can be provided solely by the electrical power steering. The second stage, based on a power balance comparison between electrical and hydraulic steering, is the decision as to which steering mode to adopt.

3 MODELLING

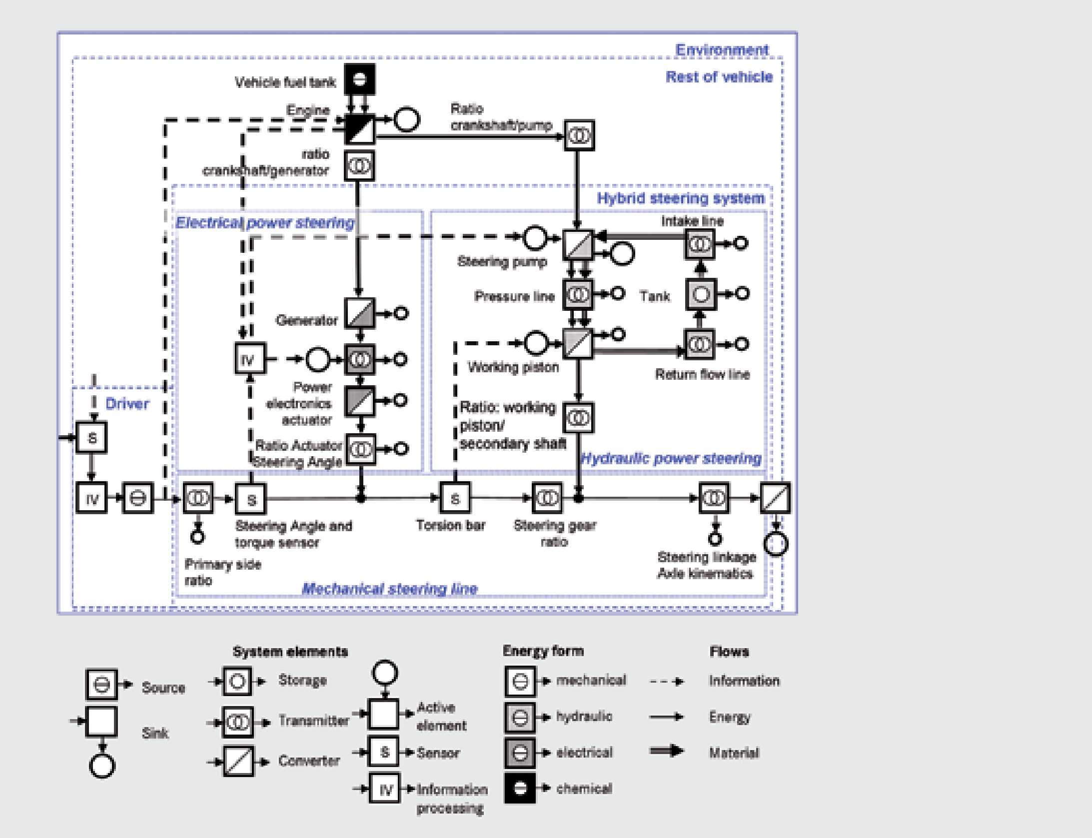
So that the choice of the optimum system design, as regards its hardware, is aligned most effectively to maximum system performance, it is essential to have a detailed understanding of the interaction between the individual parameters related to specific operating conditions, model series and system configurations. In order to be able to analyse in detail the complexity of this mechatronic system



❹ Power-controlled operation strategy of the consumption-reduced steering



❺ Simulation environment



❻ Mode of action for the hybrid steering system

and to be able to predict the power reduction potential of system configurations based on different hardware, a methodology for modelling and validating the hybrid steering system was developed as part of the development project.

3.1 SIMULATION ENVIRONMENT

A simulation model environment consisting of three tiered sub-models was designed, ❺. In the vehicle model, based on actual test drive measurement data of the vehicle speed and steering wheel angle variables, the resultant aligning torque at the steering gear output was calculated; and the servo assistance torque needed was calculated in the steering force model. Modelling the aligning torque made it possible during the development project to use the load spectra from series production vehicles, which were not fitted with their own steering force sensors. The required servo torque ascertained from the steering force model together with the engine speed, were used as input variables for the reduced consumption steering sub-model. And in that sub-model, in accordance with the operation strategy, the power balance for electrical and hydraulic steering was calculated, taking particular account of the parameters related to specific operating conditions, model series and system configurations.

Whereas, for the vehicle model, use could be made of an already validated single track model taken from previous projects, a methodology for modelling and validating both the steering force model and the reduced consumption of different steering system configurations was designed as part of the development project.

3.2 STRATEGY OF MODELLING AND VALIDATION

In close collaboration with the Institute of Vehicle Science and Mobile Machines at Karlsruhe Institute of Technology (KIT), Daimler developed a methodology by means of which a simulation environment was created, which was not only tailored to the development task as regards its efficacy but which also ensured the required overall model quality. The aim was to systematise the development process to the extent that the scope and nature of the modelling and the degree of detail pursued were systematically developed and a definitive validation path was specified. At the same time, previous knowledge of these matters and any experimental test facilities already present within the development department were to be incorporated into the modelling and validation methodology.

The modelling and validation strategy was made up of several methodological stages, which were implemented during the development project into a reduced consumption steering system and which are described in the following.

The overall system was subjected to system abstraction and system analysis and deconstructed into its sub-systems and their components; mathematical physical descriptions were formulated and the complete range of influencing parameters identified. By linking the separate component descriptions to sub-systems and to an overall system, a mathematical physical description of the power balance for the whole system was eventually produced. ❻ illustrates the effective mechatronic structure, based on [5], showing the power and energy flow of the hybrid steering system, depending on the type of energy. The system boundary between the steering system and the rest of the vehicle was chosen so that the power needed for hydraulic and electrical steering was related in each case to the mechanical energy of the power steering pump and generator in order that the power required for both systems should be comparable.

At the heart of the methodology was the establishment of a central modelling and validation matrix and of an optimisation algorithm, which could be used to calculate the combination requiring the least modelling and validation effort to be expended in order to achieve a specified overall model quality, ❼. The matrix as a whole is made up of individual modelling and validation stages for the individual system parameters of the hybrid steering system and shows the estimated achievable modelling quality (G), the time and cost involved (A), the type of modelling (MA) and the type of validation (VA). The system parameters were set out in this way so that, in accordance with a sensitivity analysis, which was conducted beforehand, the influence of the parameters on the total energy balance diminishes towards the right-hand side of the matrix and the modelling and validation accuracy increases towards the bottom of the matrix. In contrast to what may be thought of as an essentially intuitive methodology, whereby a development engineer is at pains to model in great detail those parameters, which exert a strong influence and to model to a lower quality those, which exert less influence, an optimisation algorithm calculates, for any specified model quality, that modelling and validation combination, which involves the least time and cost.

Taking the modelling of the hydraulic control valve characteristic as an example, the simplest modelling and validation accuracy is obtained by illustrating the steering gear pressure in relation to the steering gear input torque as a steady-state characteristic curve, using data provided from the supplier's service outlets. The next modelling stage is to measure the steering gear on the already existing demonstrator vehicle, without spending any more time or money on the test environment, taking the steering wheel velocity into account. The last modelling stage is taking measurements under laboratory conditions on a newly-constructed dynamic steering gear test stand, taking the effect of temperature into account. The coloured areas in ⑦ show the combination involving the least overall expenditure for a 90 % steering model quality. It seems that there is no need to construct a separate dynamic steering gear test stand just for validating the relatively strong influence exerted by the control valve characteristic on the overall power balance although this will be to the detriment of greater modelling depth for less influential parameters.

❽ illustrates the modelling and validation effort in relation to the specified modelling quality. When assessing the increase in development effectiveness while, at the same time, ensuring model quality, one point of reference used was the most costly and time-consuming modelling and validation method available. As another reference, an ambitious junior engineer was entrusted as part of a genuine trial with the task of planning a modelling and validation meth-

❼ Modelling and validation matrix: The coloured shaded boxes represent the combination involving the least expenditure for a 90-% steering model quality

Decreasing effect of the parameter

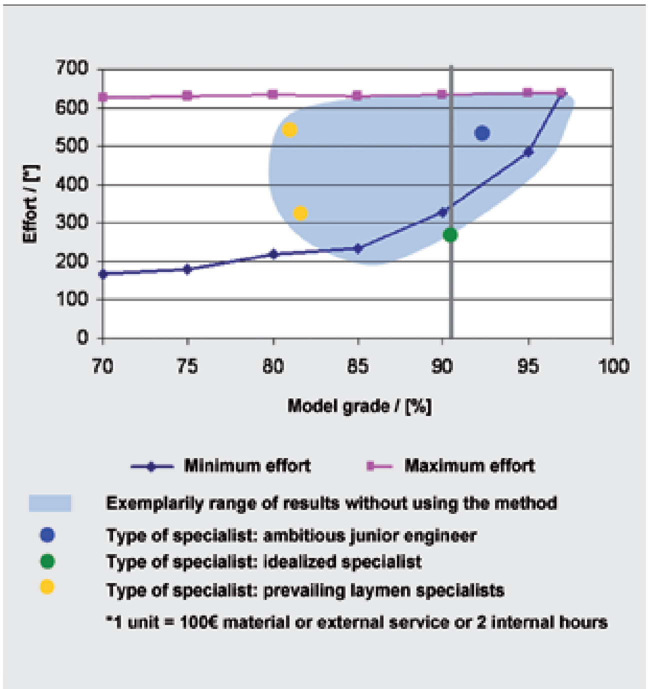
Increasing modeling depth

		Efficiency factor pump η_{LHP}	Steering sector shaft r_{SW}	Engine speed n_{Mot}	Absorption volume pump $V_{Schluck}$	Efficiency factor steering gearbox η_{LG}	Hydraulic control valve characteristic $P_{Servo,hydr}$	Maximum system volume flow Q_{sys}	Hose line radius r_i	Efficiency factor Generator η_{LM}	Circulation losses pump P_{UV}	Electrical power assistance Characteristic $P_{Servo,el}$	Density oil $\rho_{Öl}$	Control valve characteristic Δp_{SV}	Viscosity oil $\nu_{Öl}$	Hose lenght L	Pressure loss, oil filter $\Delta p_{Öf}$	Bend factor ζ_K
1	MA VA G[%] A[-]	1 4 60 0,5	1 4 90 0,5	1 4 20 0,5	1 4 76 0,5	MA VA Grade Effort	2 4 80 0,5	Characteristic Assignment from knowledge databases % Evaluation and Implementation in the simulation model							1 4 25 0,5	1 4 50 1	1 4 50 0,5	1 4 50 2,5
2	MA VA G[%] A[-]	2 4 75 1	2 4 99,9 3,5	1* 3 99,9 1	2 1 80 2,5	MA VA Grade Effort	3 3 98 5	Multidimensional mapping Measuring of the existing test vehicle % Experimentation effort Evaluation and Implementation in the simulation model							2 4 95 5	1 3 90 1,5	2 1 90 9	1 4 80 3
3	MA VA G[%] A[-]	3 4 80 1,5			3 1 90 4,5	MA VA Grade Effort	3 2 99 85	Multidimensional mapping Measuring at the newly established testing configuration % New Construction test rig Operating costs + transaction of the measurement Evaluation and Implementation in the simulation model							2 4 98 6	1 4 99,9 4,5	2 4 97 1,5	1 4 85 3,5
4	MA VA G[%] A[-]	3 1 99,2 55			3 1 99 55			99 55	99,9 4,5	99,9 2	70 7	96 9,5		95 3				1 4 97 6
5	MA VA G[%] A[-]										3 1 99,2 46	2 2 97,5 79		3 2 99,3 79				1 1 99,8 10

In addition an effort of 62 effort units has to be made for preparation of the modeling and validation matrix

Modeling type (MA): 1: Constant, 2: Characteristic, 3 : Multidimensional mapping; 4 : other sub-model
Validation (VA): 1: Ancillary component test stand, 2 : Steering gear test stand, 3 : Demonstrator, 4 : Knowledge databases
Colored shaded boxes: Best combination of modeling and validation strategy for a 90% validation

❽ Result of the modeling and validation strategy combination involving the least expenditure for a 90-% steering model quality



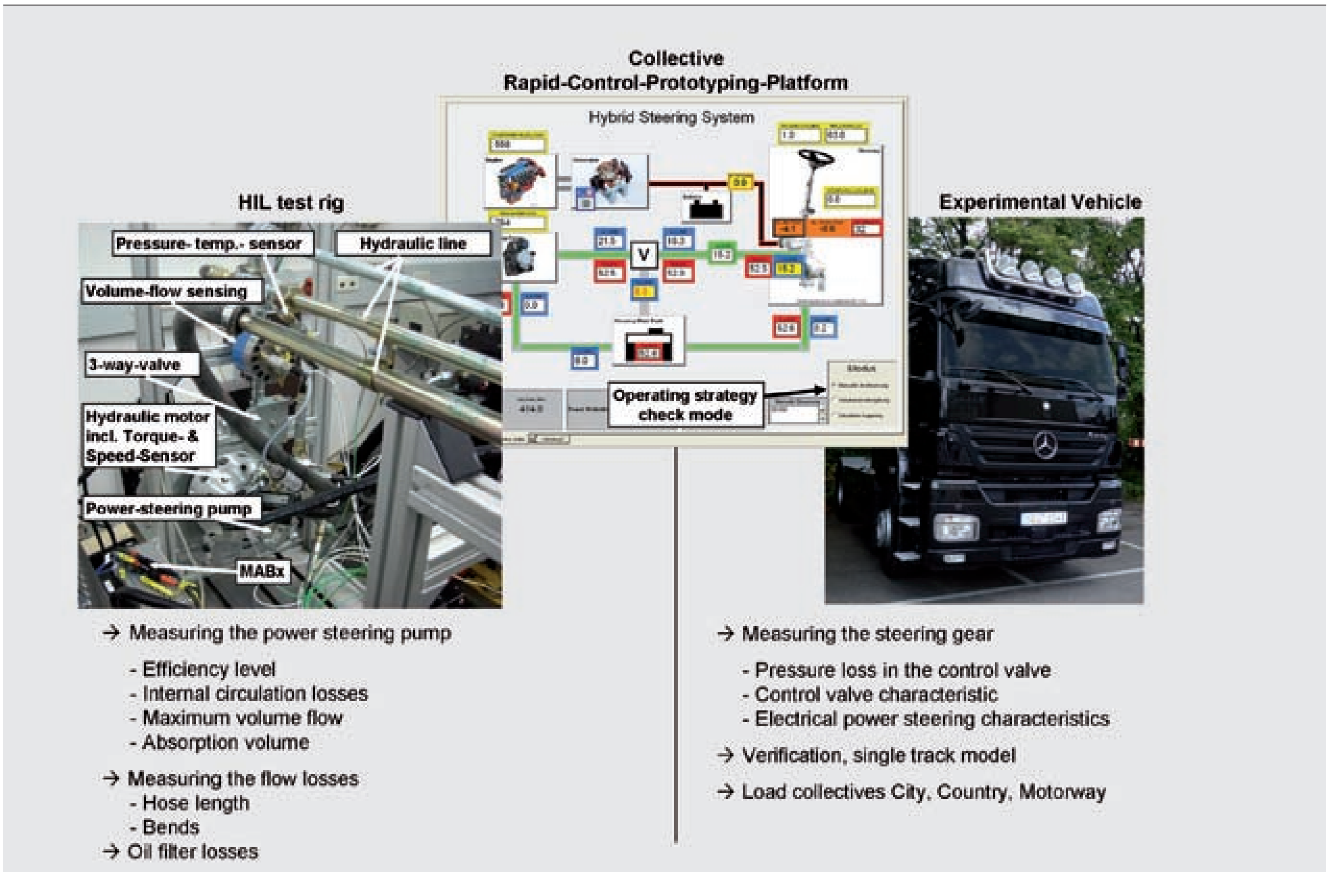
odology for a 90 % model quality without using an optimisation algorithm. This engineer achieved a modelling quality, which, although 3 % better than the specified 90 %, cost 520 points or 58 % more than the optimised case. Compared with a notionally ideal expert who, without the time and expense involved in drawing up the modelling and validation strategy, immediately chose the optimum combination for achieving the required model quality at the least cost, the method is highly dependent on the individual's level of expertise. The method is characterised by systematising the modelling process so that modelling-quality uncertainties are reduced and the cost and effort of modelling is aligned to the least development cost.

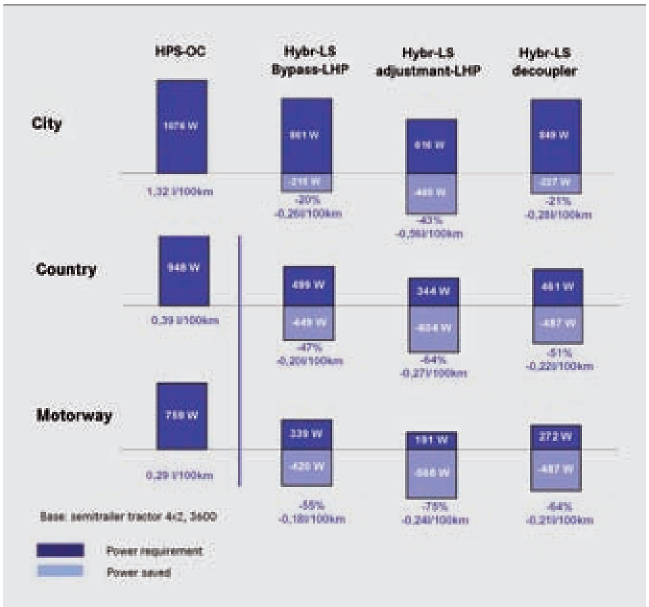
4 EXPERIMENTAL VALIDATION AND SIMULATION RESULTS

As specified by the modelling and validation strategy, the model was validated on a test vehicle and on an HIL test stand, after which the results were reproduced in the simulation environment. ❾ illustrates the experimental test environment.

An Mercedes-Benz Axor 1840 4x2 semitrailer truck was used as the test vehicle with a hybrid steering system consisting, at this early development stage, of a purpose-built steering adjustor with an Emoteque permanently excited synchronous motor coupled with

❾ Test environment to validate the simulation model





➓ Power reduction potentials and resulting fuel consumption reduction for different hybrid versions

SWM-50 power electronics supplied by Maccon [6]. As development progressed, the actuator was replaced by the "Servotwin" hybrid steering system manufactured by ZF-Lenksysteme GmbH [7]. An EV2 power steering pump supplied by Ixetic-GmbH was used as the active volume flow adjustment mechanism, allowing the system volume flow to be restricted by a controllable valve [8] in the bypass. In addition, the hydraulic steering system was fitted with pressure and volume sensors and the electrical sub-system equipped with current and voltage sensors.

The HIL test stand environment was based on an existing auxiliary assembly test stand set up in the main Daimler AG research department and supplemented, for taking these measurements, by the individual hydraulic steering components.

Both the performance-led operation strategy for the reduced consumption steering system and the test stand controls were implemented in Matlab/Simulink. The test environments and the operation strategy prototypes were managed in a common rapid control prototyping platform supplied by the company dSpace.

With the aim of identifying an optimum system design during the early development phase and of taking subsequent development work in the direction of the greatest potential, different hardware configurations for the active volume flow adjustment mechanism were examined. The system configurations examined by simulation were a bypass pump, in which the excess system volume flow was recirculated via a return run inside the pump, an adjustable pump with variable absorption volume and a conventional power steering pump linked to a binary switchable clutch. ➓ summarises the simulation results.

The results show that, compared with the other two system configurations, a controllable adjustable pump possesses the greatest power reduction potential of 60 up to 75 %. However, the active adjustable pump's advantage in offering the greatest power reduction potential has the disadvantage of requiring more space to install it. A controllable bypass pump, on the other hand, represents an attractive solution, especially for long-distance applications, not least because it is easy to install. Although a clutch has a slightly greater power reduction potential compared with the bypass pump, it not only requires more installation space and poses design challenges as regards robustness but, because it employs binary switching, it places high demands on its control mechanism.

5 SUMMARY

A Daimler AG development project undertaken in close collaboration with Karlsruhe Institute of Technology (KIT) employed modelling to design a reduced fuel consumption hybrid steering system. When designing the reduced consumption steering system and its simulation, a method was used, which allowed the modelling and validation process to be systematised to the extent that uncertainties as regards achieving the required model quality were minimised and an effective development methodology was chosen, which took account of existing development resources and previous knowledge.

A validated simulation environment predicted the fuel reduction potential of different system configurations. It was demonstrated that a power reduction potential of 60 to 75 % could be achieved compared with the conventional steering system installed in commercial vehicles.

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